Investigation of optimal design and operation of a small-scale Low Temperature District Heating (LTDH) with multiple heat sources

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Abstract

Improving energy efficiency and reducing carbon emission is among the immediate measures being taken in many countries to address the challenge of climate change. Energy consumption in buildings accounts for over 40% of total primary energy demand in the European Union (EU). Most of this energy is in the form of heat for space heating in buildings which is commonly supplied in building using onsite fossil fuelled-boiler installations in EU. The current fossil fuelled boilers are designed to supply heat at high temperature (about 80 °C) and usually oversized for the required load capacity. This process suffers from low overall thermal efficiency of the heat supply systems.

In this project, it was sought to investigate an integrated approach of supplying heat to buildings by aggregating various types of heat sources and delivering heat to a common heat distribution network to form a small-scale district heating system. This is considered as an effective solution to increase efficiency through lowering the hot water temperature and encouraging the adoption of renewable energy systems. Therefore, this thesis investigates the operation and design optimisation of a Low Temperature District Heating (LTDH) network with multiple heat feed-in sources such as a heat pump, biomass boiler, gas boiler and solar thermal collector. A case study to evaluate the design of system was considered as part of the Creative Energy Homes (CEHs) at the University of Nottingham. An overall heat load demand of the site was evaluated using Energy Plus software and a computer model for low temperature heat with multiple heat sources was introduced to optimise different feed-in heat sources. To improve heat provision flexibility, maximise heat generation from renewable sources and provide heat networks flexibility, an optimisation model of the thermal store was also carried out. Furthermore, this work investigated the environmental and economic viability of the proposed low temperature heat network.

The case study involves the Creative Energy Homes which consists of seven low energy homes with an aggregate heat load of 44 kW and annual energy consumption of

40258.1 kWh, including 14110.89 kWh for domestic hot water and 26417.92 kWh for space heating. It was established that a system consisting a 1.56 kW solar collector,10 kW heat pump, 15 kW biomass boiler, 20 kW gas boiler and a thermal store of 0.894 m^3 (or 25.96 kWh) can supply heat to the site using a LTDH system at the lowest cost and with the least environmental impact. The system's annual operation cost and carbon emission was £ 1997.87 and 1634.4 kg respectively. It was also found that the biomass boiler and heat pump supplied more than 80% of heat demand of the site, while the gas boiler fulfilled less than 10% of heat demand, working as auxiliary boiler. The solar collector operated for a total of 1891 hours per year and contributed less than 10% of the heat demand.

Acknowledgments

I am very thankful to my principal supervisor Dr. Rabah Bouhkanouf who has always been available to provide guidance, support and encouragement throughout all stages of my PhD.

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Finally, I would like to dedicate this thesis to my parents and my family for their endless love, unconditional support, encouragement and sacrifice. To Wenfei Liu, who represents my girlfriend, thanks for her support, love, encouragement and sacrifice.

Contents

List of Figures

List of Table

List of Acronyms

- AOT: Advanced Oxidation Technologies
- ASHP: Air Source Heat Pump
- BPS: Building Performance Simulation
- CCI: Climate Change Initiative
- CMEMS: Copernicus Marine Service
- CEHs: Creative Energy Homes
- CFC: Chlorofluorocarbons
- CHP: Combined Heat and Power
- DECC: Department of Energy and Climate Change
- DH: District Heating System
- DSCHP: Doubled-Stage Coupled Heat Pumps
- ECO: Energy Company Obligation
- EPC: Energy Performance Certificate
- ERA5: European Centre for Medium-range Weather Forecasts
- ESA: European Space Agency
- EUDP: Energy Technology Development and Demonstration Programme
- GCHP: Ground Coupled Heat Pump
- GISTEMP: National Aeronautics and Space Administration Goddard Institute for Space Studies
- GSHP: Ground Source Heat Pump
- GWHP: Ground Water Heat Pump
- HadCRUT: Hadley Centre and Climatic Research Unit
- HNDU: Heat Networks Deliver Unit
- HNIP: Heat Networks Investment project
- HDPE: High Density Polyethylene
- HIU: Heat Interface Units
- IES VE: Integrated Environmental Solutions Virtual Environment
- INDC: Intended Nationally Determined Contribution
- JRA-55: Japan Meteorological Agency
- LTDH: Low Temperature District Heating
- NASHP: Non-frosting Air Source Heat Pump
- NOAA: National Oceanic and Atmospheric Administration
- PB: Polybutylene

- PET: Polyethylene Terephthalate
- PEX: Cross-linked Polyethylene
- PUR: Polyurethane
- RCP: Rapid Crack Propagation
- RHI: Renewable Heat Incentive
- RO: Renewables Obligation
- SAGHPS: Solar Assisted Ground Source Heat Pump System
- SSPHS: Small-Scale Pellet Heating System
- SWHP: Surface Water Heat Pump
- TES: Thermal energy storage
- VIP: Vacuum Insulation Panels
- WTO: World Health Organization

Nomenclature

- A: surface area of thermal store (m^2)
- C_P: specific heat capacity of water [$kJ/(kg \cdot K)$]
- d: diameter of thermal store (m)
- E_b: output of biomass boiler (kW)
- f_b: cost of fuel (£/kg)
- f_{b-gs}: biomass boiler tariff of government subsidy (£/kWh)
- $f_{elec-day}$: standard electricity price (£/kWh)
- $f_{elec-night}$: peak-off electricity price (£/kWh)
- f_g: gas price (£/kWh)
- f_{h-gs}: heat pump tariff of government subsidy (£/kWh)
- f_{s-gs}: solar collector tariff of government subsidy (£/kWh)
- $f_{taxtation}$: total carbon emission taxation per kilogram (£/kg)
- f_{th}: maintenance cost per kilowatt hour (£/h)
- $F_{biomass}$: annual cost for fuel (£)
- F_{b-m}: annual maintenance for solar collector (£)
- $F_{electrcity}$: the annual cost of electricity (£)
- F_{gas}: annual cost for gas (£)
- F_S: annual operation cost for solar collector (£)

- F_{s-gs}: annual government subsidies for solar collector (£)
- $F_{taxtation}$: total carbon emission taxation from all the heat sources (£)
- F_{th}: the total thermal storage maintenance cost (£)
- G: solar radiation on solar collector (W/m^2)
- h: height of thermal store (m)
- i: day
- j: hour
- LHV_b: Lower Heating Value of fuel (kWh/kg)
- m_{b1}: mass of fuel for working period(kg)
- m_{b2}: mass of fuel for star-up period (kg)
- m_{b-cw}: mass of internal water in biomass boiler (kg)
- $m_{cw-d,i}$: hourly volume flow rate (m^3/h).
- m_{cw-max}: mass flow rate of water pump (kg/h)
- m_{g-cw}: mass of internal water in gas boiler (kg)
- m_B: carbon emission per kilowatt hour for biomass boiler (kg/kWh)
- m_G: carbon emission per kilowatt hour for gas boiler (kg/kWh)
- m_H: carbon emission per kilowatt hour for heat pump (kg/kWh)
- M_B: annual carbon emission from biomass boiler (kg)
- M_G: annual carbon emission from gas boiler (kg)
- M_H: annual carbon emission from heat pump (kg)
- M_{total}: total carbon emission of all the heat sources (kg)
- $P_{elec-d,i}$: hourly power input (kW)
- P_{sh}: heat production of solar collector (W)
- q_{th-t}: hourly thermal storage capacity (kWh)
- $Q_{b-\mathrm{out}}$: annual heat produced by biomass boiler (kWh)
- Q_e: electricity input(kW)
- Q_{b-cw}: heat energy for internal water circuit of biomass boiler (kWh)
- Q_{g-cw}: total energy for internal water circulation of gas boiler (kWh)

- $Q_{g-input}$: annul input energy of gas (kWh)
- Q_{g-out}: annual heat produced by gas boiler (kWh)
- Q_{h-out}: annual heat produced by heat pump (kWh)
- $Q_{load-max}$: maximum heat load (kW)
- Q_p: heat pump output (kW)
- Q_{s-big}: maximum energy stored in thermal store (kWh)
- Q_{s-out}: annual heat produced by solar collector (kWh)
- Q^B: heat supplied by biomass boiler (kW)
- Q^G: heat supplied by gas boiler (kW)
- Q^H: heat supplied by heat pump (kW)
- Q^{In}: the accumulation of heat supply by multiple heat sources (kW)
- Q^{Intial}: initial thermal store or the minimum thermal store capacity (kW)
- Q^{loss}: heat loss of thermal store (kW)
- Q^{Out}: heat load for the dwellings
- Q^S: heat supplied by solar collector (kW)
- $Q^{Storage}$: capacity of thermal store (kW)
- S: solar collector area (m^2)
- T_a: ambient temperature (°C)
- T_f: collector average temperature (°C)
- T_{in}: average temperature of water inside the thermal store (°C)
- T_R: return water temperature (°C)
- T_S: supply water temperature (°C)
- U: overall heat transfer coefficient ($W/(m^2K)$)
- V: volume of thermal store (L)

Symbols

- α: working on/off of the heat pump
- α_1: the first-order coefficient in collector efficiency equation, [$W/(m^2$°C)]
- α_2: the second-order coefficient in collector efficiency equation, [$W/(m^2$°C)]
- β: working on/off of the biomass boiler
- γ: working on/off of the gas boiler

- Δt_{b1}: the total biomass boiler working hour (h)
- Δt_{b2}: the number of stat-up of biomass boiler
- Δt_{b-d}: total working hour during daytime for biomass boiler (h)
- Δt_{b-n}: total working hour during night time for biomass boiler (h)
- Δt_{h-d}: heat pump working hour during day time (h)
- Δt_{h-n}: heat pump working hour during night time (h)
- ΔT: the temperature difference between the water temperature and ambient temperature (℃)
- ΔT_s: temperature difference for thermal store supply/ return temperature (℃)
- η: intercept (maximum) of the collector efficiency.
- η_b: the efficiency of biomass boiler
- η_g: the efficiency of gas boiler
- ρ: the density of water (kg/m^3)

1 Introduction

1.1 Background

The background of this research project is based on fulfilling the energy consumption without fossil fuel in the long term and ensuring energy security under the premise of protecting the environment and improving the quality of people's life. It is common established that energy consumption and standards of living are strongly correlated. In the developed countries energy consumption per capita is high which reflects the high standard of living compared to that in developing countries. The high living standards aspired by the developing countries however is fuelling the ever increase in the amount of burnt fossil fuels in all forms, as shown in Figure 1.1 where most of energy consumption increase is recorded in the Asia Pacific countries. It can also be seen that the total global energy consumption in 2018 increased by 18.45% compared to that in 2008.

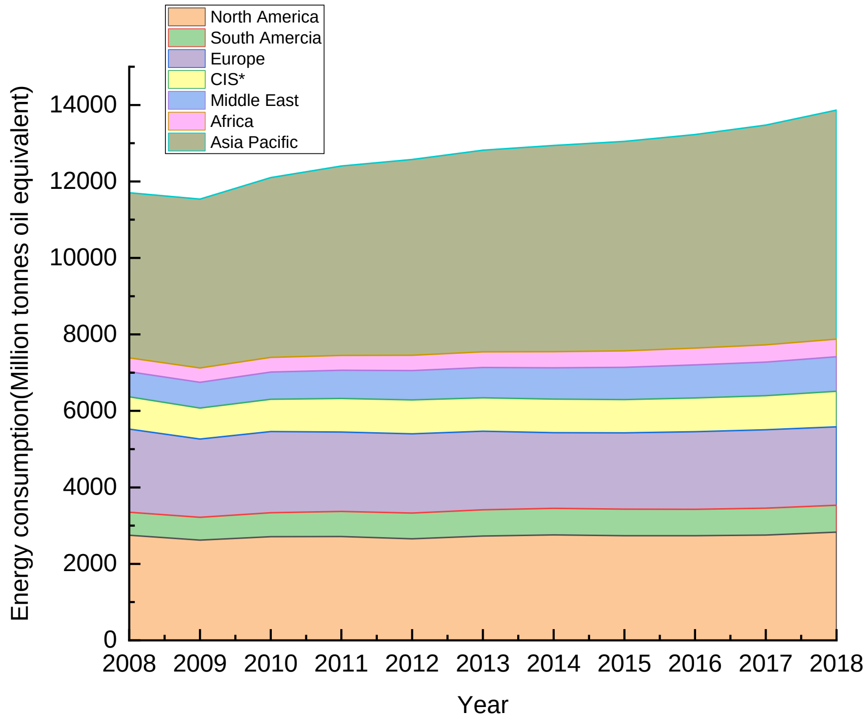


Figure 1.1 Primary energy consumption in each region from 2008 to 2018 [1]

The rate of primary energy consumption is though not evenly distributed as most the current increase in fossil fuel consumption is natural gas. This is particularly seen as a short to medium term substitute to a more polluting fuels such as coal. Figure 1.2 shows the percentage of global primary energy consumption by fuel. It can also be seen that there is a marked decrease in energy consumption from coal and oil while little change in nuclear and hydroelectricity energy generation and an increase in renewable energy sources still forms a small fraction of the total consumption.

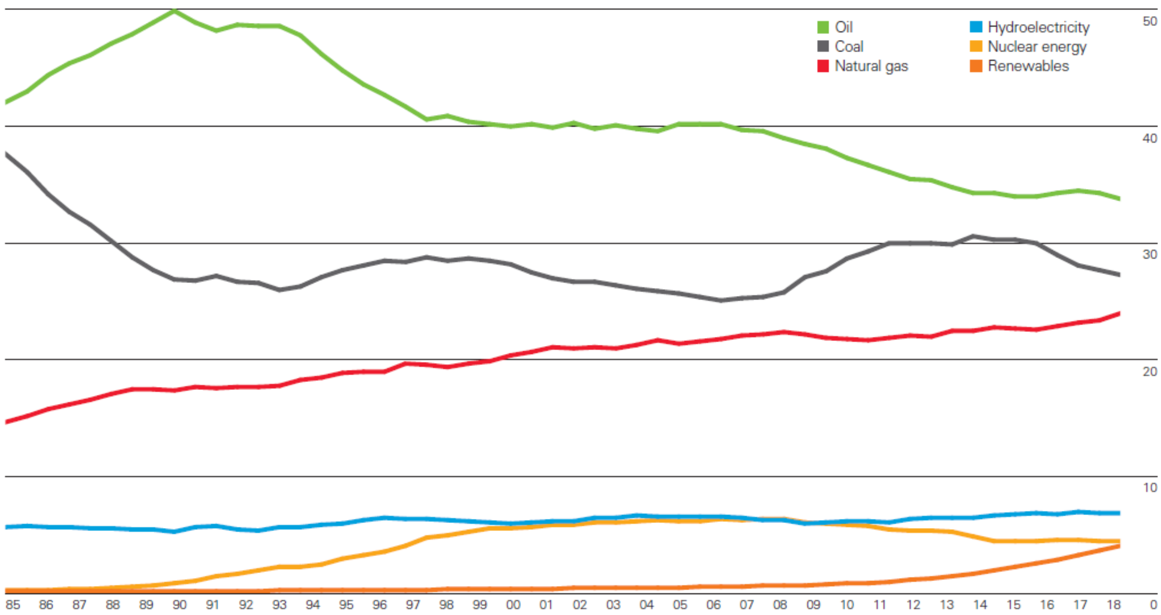


Figure 1.2 The percentage of global primary energy consumption by fuel [1]

The global CO_2 emissions presented a similar trend to energy consumption, which increased by 131.7% in 2016 compared with that in 1975. The CO_2 emissions increased significantly due to the high demand for the fossil fuel. Coal and oil contributed most of CO_2 emissions as Figure 1.4 shows, the CO_2 emission from coal and natural gas is still increasing. Despite extraordinary growth in renewable energy, fossil fuels still dominate the global energy system.

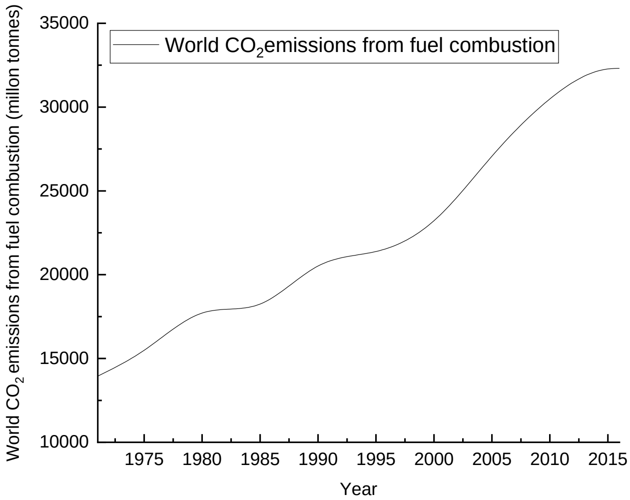


Figure 1.3 Global emissions trend from fuel combustion [2]

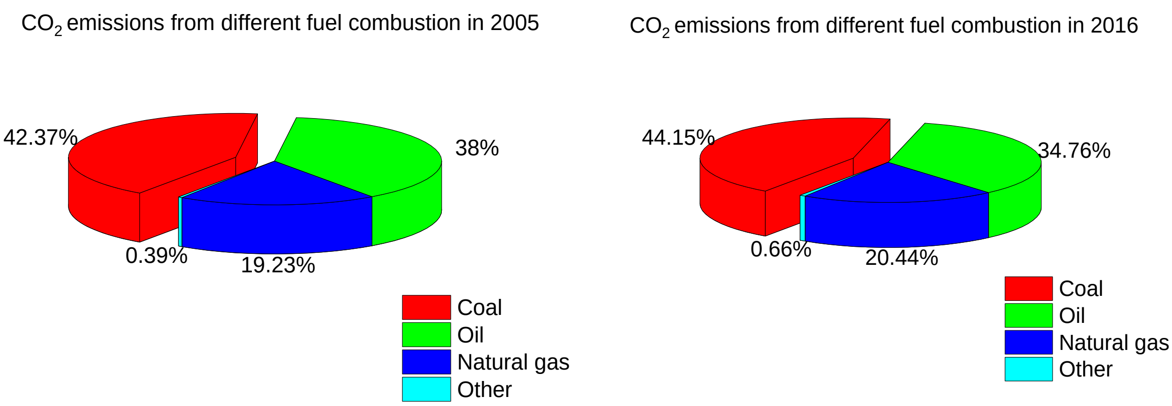
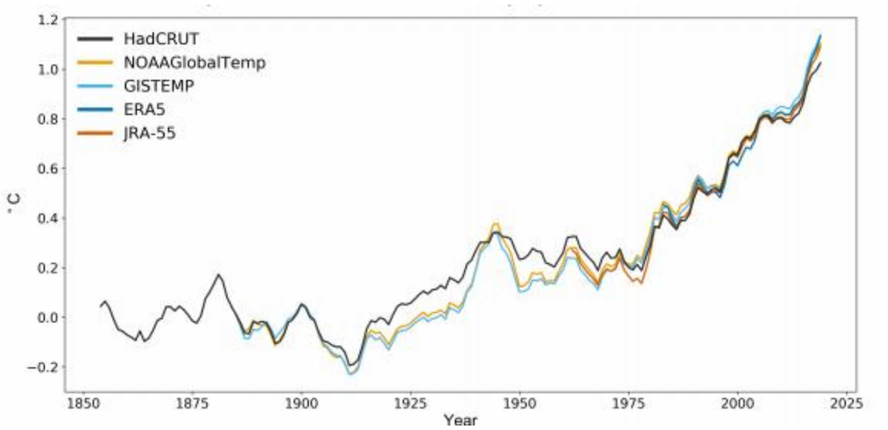


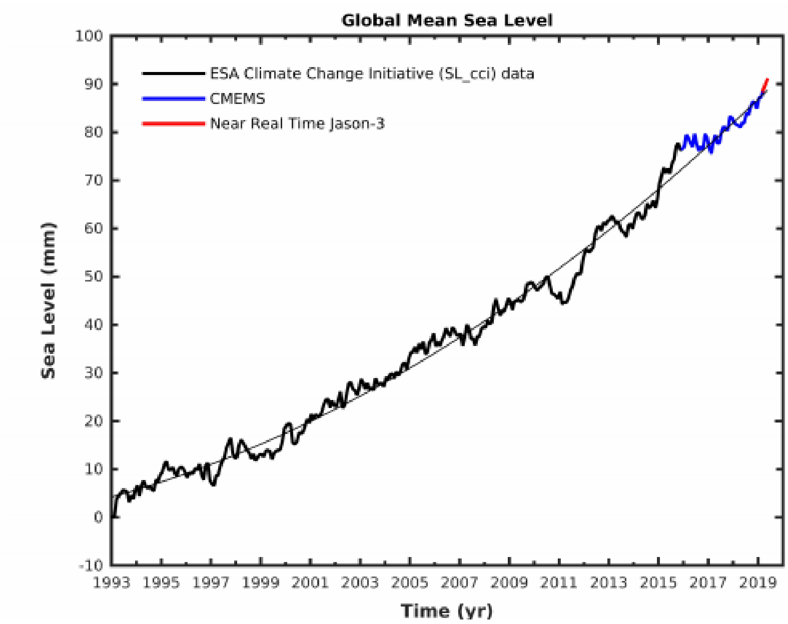
Figure 1.4 CO_2 emissions from different fuels combustion in 2005 and 2016 [2].

The increasing level of CO_2 concentration in the atmosphere can lead to many environmental problems globally. The average global temperature for 2015-2019 is on track to be the warmest of any equivalent period on record. It is currently estimated to be 1.1 °C higher than that in pre-industrial (1850-1900) times and 0.2 °C warmer than that in 2011-2015 as Figure 1.5 shows [3, 4]. In addition, the sea-level rise is accelerating due to ocean warming and land ice melting from West Antarctica and Greenland [4]. Global mean sea level rose from 3.04 millimetres per year during the period 1997–2006 to approximately 4 millimetres per year during 2007-2016 as Figure 1.6 shows [4]. 25% of annual emission of anthropogenic CO_2 was absorbed by seawater, and as a result the ocean is becoming more acidic.



*The global temperature assessment is based on five datasets: HadCRUT.4.6.0.0 (UK Met Office Hadley Centre and Climatic Research Unit, University of East Anglia), GISTEMP v4 (National Aeronautics and Space Administration Goddard Institute for Space Studies), NOAAGlobalTemp (National Oceanic and Atmospheric Administration (NOAA), ERA5 (European Centre for Medium-range Weather Forecasts), and JRA-55 (Japan Meteorological Agency).

Figure 1.5 Global mean temperature difference from 1850-2019 for five data sets [4].



* Data source: European Space Agency (ESA) Climate Change Initiative (CCI) sea-level data until December 2015, extended by data from the Copernicus Marine Service (CMEMS) as of January 2016 and near real-time Jason-3 as of April 2019

Figure 1.6 Global mean sea level from 1993-2019 [4].

It is estimated that the construction sector and associated buildings account for 36% of global final energy consumption and nearly 40% of total direct and indirect CO_2 emissions [5]. For buildings, the energy is mainly consumed for space heating, domestic hot water, cooking, lighting and appliances. The energy consumption for cooling was much less than that for heating due to the climatic conditions of northern European countries [6]. According to Figure 1.7, energy used for heating, including both space heating and domestic hot water, accounted for 81% of total energy consumption. Similarly, the emission from heating was the single biggest contributor to UK emissions, accounting for 37% of total emissions in the UK [7].

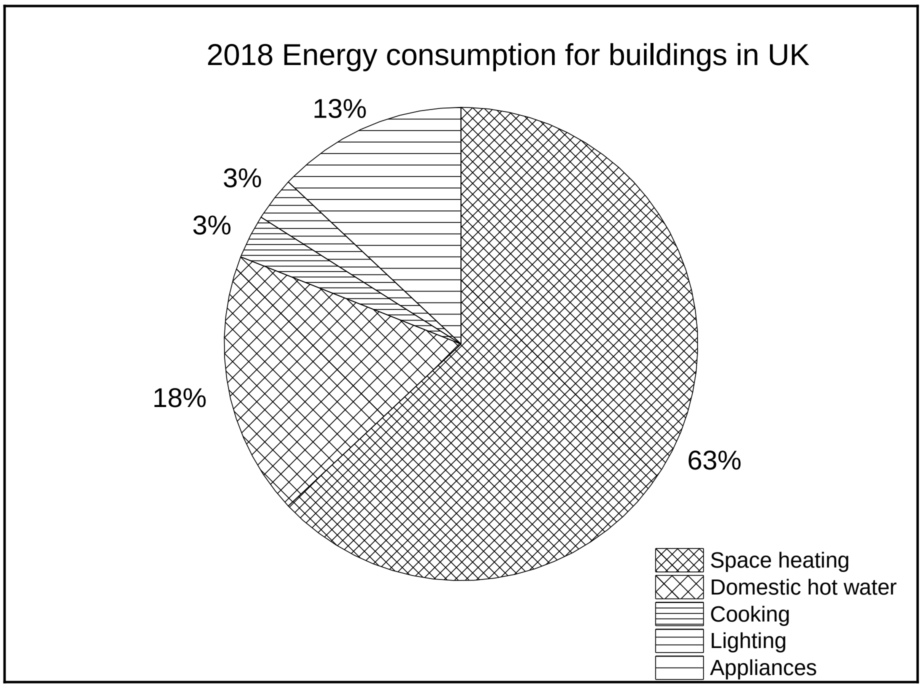


Figure 1.7 2018 Energy consumption for buildings in UK [6]

1.2 Motivation and contribution to knowledge

Heat energy consumption in the UK forms the largest energy consuming sector (see Figure 1.8), while carbon emissions from space heating and domestic hot water accounted for 37% of total emissions from heat generation. Therefore, primary energy used to generate heat in the UK is important for carbon emission reduction and energy saving. The heating generation method in domestic buildings plays an important role. 84% of residential buildings in the UK adopt gas as the main heating fuel, and the houses heated by the electricity accounted for 8.6% [8].

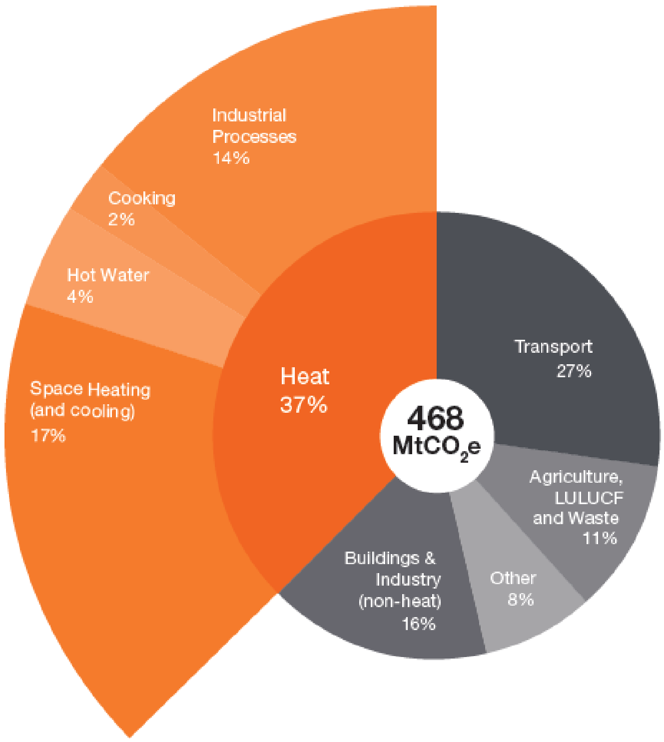


Figure 1.8 UK emission in 2016 across different sectors [9]

The UK government set a goal to reduce 80% of carbon emissions by 2050 [10]. Buildings accounted for 34% of total carbon emission in 2014 [11]. With the increase of population, Energy Saving Trust has analysed the impact of a 2050 ready policy. It is assumed 200000 homes are built each year between 2019 and 2032 (the end of the 5th carbon budget), and the number will reach 43 million by 2050 [12]. In order to achieve the objective of carbon emission reduction, the new houses should be built more energy efficient. However, 80% of houses will be occupied by 2050 have already been built. 26 million houses should be retrofitted to reduce greenhouse gas emissions [13]. In 2017, the residential sector, with the emission of 64.1 Mt, accounted for 17% of all carbon dioxide emissions [14]. So that low energy housing solutions are an essential part of achieving the Government's target, researchers at the Department of Architecture and Built Environment at the University of Nottingham were approached by construction firms seeking collaborations to help develop and test low-energy housing solutions. Creative Energy Homes were built to varying specifications to support the testing of a variety of design strategies, construction methods and technologies intended for the volume house-builder market.

Nowadays, buildings are heated by individual fossil fuel boilers and/or renewable heat sources (biomass boilers, solar collectors or heat pumps). Technically gas and biomass boilers which convert chemical energy of a fuel into heat have reached their energy performance limit under condensing mode, and as high as 95% of efficiency can be obtained. Individually installed solar thermal collectors usually suffer from high stagnation temperatures and the energy generation capacity is not fully utilised. The energy performance of heat pumps on the other hand depends on many factors including climatic conditions, design of heat extraction and rejections loop as well as heat supply temperature. Therefore, aggregating heat generated from various energy sources in a heat distribution network will maximize the use of the renewable system’s capacity, reduce fossil fuel energy consumption and mitigate carbon emission. Furthermore, supplying heat through a Low Temperature District Heat (LTDH) network will reduce energy loss associated with the heat distribution. In designing a LTDH system, many technical and economical problems need to be investigated like evaluating the heat load of buildings forming a community, optimising and integrating different heat sources (solar collector, heat pump, biomass boiler and gas boiler) into the LTDH system, sizing and selecting the thermal storage and assessing the environmental impacts and cost benefits of the system.

The current practice is that buildings in a community are fitted with individual heat source to meet the heating demand of each building. The dwellings have multiple heat sources (gas boiler, biomass boiler, solar collector and heat pump), which are designed for the peak heat load of each building. Therefore, sharing heat generation will smooth out the peak generation capacity of individual system. In this project, a research gap about optimisation of the heat sources in the LTDH system has been identified. The main contribution of the research centres on design optimisation of low temperature small-scale district heating with multiple heat sources, which includes sizing of solar collectors, biomass boiler, heat pump and gas boiler and the thermal store for a dynamic heat load of a community. The optimisation of the thermal store in LTDH is critical to storing the surplus heat from multiple heat sources particularly renewable energy and then supplying heat to buildings.

1.3 Aim and Objectives

The main aim of this thesis is to investigate the design and feasibility of Low Temperature District Heating (LTDH)system for small communities. A case study of a small community made up of seven houses was used to assess the heat networks. Firstly, the hourly heat load of the community was established with EnergyPlus software. Then, the potential of many heat sources and thermal storage were considered by developing a mathematical model to simulate the operation of the LTDH.

The main research objectives of this work can be summarised as follows:

- Evaluate the heat load of the Creative Energy Homes (CEHs) buildings as part of a small-scale community
- Assess the different heat sources capacities to identify the optimum heat sources for LTDH
- Optimise the thermal store to satisfy the site’s heat load reliably
- Analyse the economic and environmental impact of the LTDH system

1.4 Thesis outline

This thesis is structured into seven chapters and represented schematically as shown in Figure 1.9.

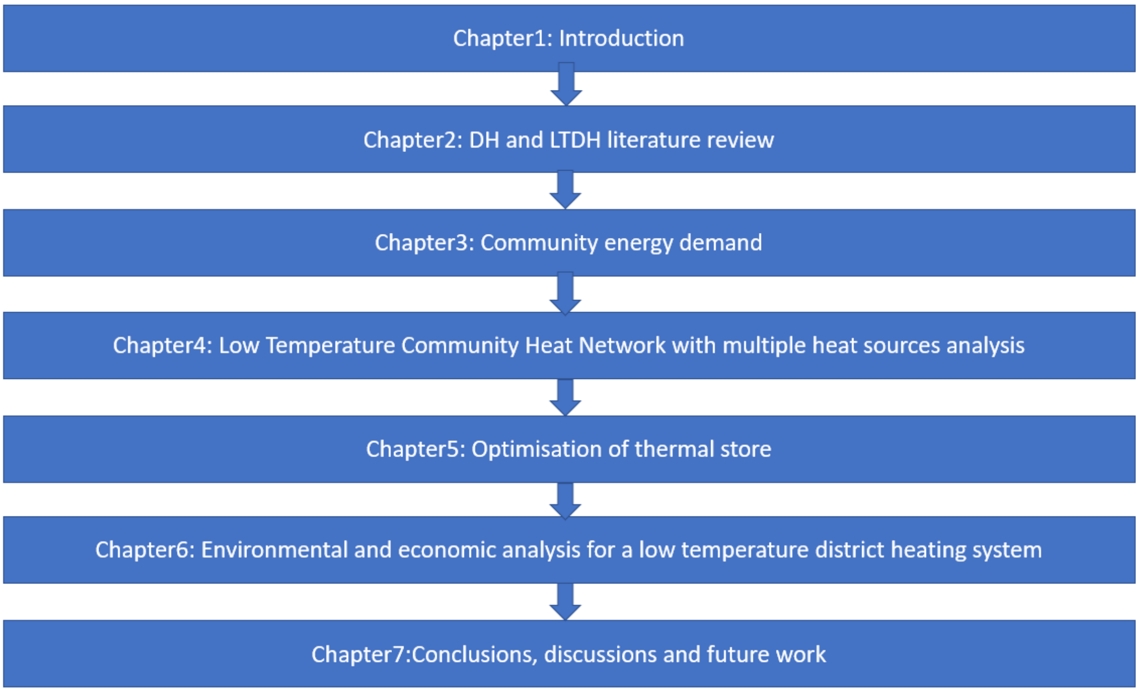


Figure 1.9 Structure of the thesis

Chapter 1 introduces the topic of the thesis through a brief background, motivation and contribution of the work and outlines the aims and objectives of the research project.

Chapter 2 reviews the DH and LTDH systems, illustrating their potential role in the transition towards low carbon buildings. Firstly, the review of the elements in the heating system is summarised, including heat sources, heat networks and supply. Then, the district heating technology is described by the trend of district heating and the current market of district heating. It is also pointed out the evolving technology of new generation of DH, highlighting its potentials to fully exploit the renewables and low carbon heat sources due to the operation with low temperatures. In addition, the challenges of operating existing heat networks, supporting policies and outlook for district heating are discussed.

Chapter 3 introduces the case study site including the construction method, efficient heating and cooling system. An Energy Plus computer model for the heat demand of the buildings in the site is performed and aggregate heat demand profiles are established.

Chapter 4 introduces the method to optimise the heat sources in the LTDH system. The methodology is applied to different scenarios, and the thermal store capacity is the criterion to design and optimise heat sources.

Chapter 5 introduces the method to optimise the thermal storage in the LTDH system. The methodology is applied to different scenarios based on the optimum heat sources in Chapter 4. The thermal store size is determined by the storage capacity. The thermal store can be optimised by adding the heat loss. The results for the optimum heat generation and storage capacity with three size solar collectors are obtained.

Chapter 6 analyses the environmental and economic aspects of the case study of the LTDH system in which various design scenarios are taken into account, as discussed in Chapter 5. The environmental and economic impacts involve the evaluation of the system cost and carbon emission.

Chapter 7 concludes this thesis critically and reflects on the results obtained, recommendations and future work development of LTDH.

2 Review of District Heating (DH) system

2.1 Introduction

This chapter begins with a review of potential heat sources for integration into a district heating system such as fossil fuelled boilers, heat pump, solar thermal collector, and Combined Heat and power (CHP). The renewable energy is especially discussed from the aspects of solar collectors, geothermal heat and biomass energy. The category of heat interface unit and heat emitter are described in detail. The different types of networks are described and a comparison of disadvantages and advantages is carried out. In addition, the heat losses and diameters are summarised.

Subsequently, the district heating system and the market for the district heating system in the world, like China, the USA, Russia and the EU countries are generally introduced. The trend of district heating over the past years has been described and the direction for district heating turns to renewable heating system with more energy saving and higher efficiency are presented. The development of district heating system in the UK in the past years is especially depicted.

The next one is a description of the concept of Low Temperature District Heating system. The present work in the field is defined through the review of current LTDH technology, focusing on the implementation of space heating demand and domestic hot water supply for low energy houses by the LTDH. The barriers of integrating LTDH system into the existing buildings are described, including the Legionella risk and each element in the heating source preparation for applying LTDH [15].

Then, the policies from different countries to decrease energy consumption and achieve decarbonisation are presented, and the challenges for district heating in different countries are summarised. Meanwhile, the future direction for district heating in each country is also predicted.

2.2 Heat sources for district heating

District heating refers to the heat from one or several heat sources to the dwellings and buildings through networks [16]. The district heating technologies vary widely among countries, depending on the following aspects [17]:

- Energy security
- Energy policy of each country
- The reliance of own fuel resource
- Regulation
- Climatic and local conditions
- Economic development
- Access to new and innovative technologies

Although the characteristics, technologies and geometries of the district heating system are different, three major aspects get involved: the heat source, network, and end-user. There is a heating centre including one or multiple heat sources, which balances the heat demand and provides backup/peak supply as Figure 2.1 shows [17, 18]. Then the hot water from the heat generation plant is distributed and transported to the secondary network via the primary network, while substations connect the primary and secondary networks. Finally, the water is transported to the end-user via a secondary network.

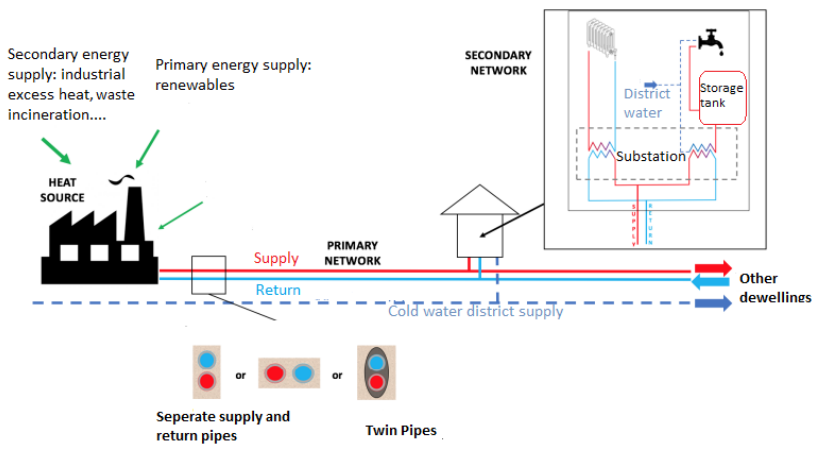


Figure 2.1 Layout of a district heating network with indirect space heating and hot water supply [18]

Heat sources for the district heating system can be divided into CHP system and heat-only system according to the function. The most common energy used as a heat source is fossil fuels, namely oil, coal and gas. Renewable energy source, such as biomass and solar energy, has attracted more attention in recent years.

2.2.1 CHP

CHP technology is electricity generation equipment, where the waste heat is also generated for heating and cooling. CHP can be divided into large-scale CHP, small-scale CHP, and micro-CHP according to the output power. Table 2.1 shows the types of CHP plants with electrical outputs, engines and applications.

Table 2.1 The types of CHP plants [19].

Types of CHP plant	Electrical outputs	Engines	Application
Large-scale CHP	>2MW	Gas turbines Reciprocating engines	Large industrial site
Small-scale CHP	<2MW	Spark ignition engines Micro-turbines Small scale gas turbines	Small industrial site Buildings and community
Micro-CHP	<50kW	Stirling engines	Domestic and small commercial site

Heat engines used to provide main power to produce electricity and heat can be categorised into gas turbine, steam turbine, reciprocating engine and combined cycle [20]. Fuels, which are used in CHP, can be divided into commercial fuels (coal, fuel oils, gas oils and natural gas) and waste fuels (solid waste fuel, liquid waste fuel and gaseous waste fuel). Compared with other CHPs, the ratio of power to heat in the gas turbine is the highest, ranging from 0.5 to 2. In addition, hot water and steam can be produced by the gas turbine due to exhaust. However, gas turbines are used to satisfy the peak load of electricity due to gas prices. The gas engines can be divided into reciprocating gas engine and spark- ignition engine. The former is used in small DH applications, low temperature and individual building [21]. Spark-ignition engines compose base heat load and heat storage is needed to maximize the production of electricity. The efficiency of electricity production used by the fuel cell ranges from 30% to 60% with the power to ratio of 0.5-1.4. Combined with other technologies such as gas and steam turbines, fuel cells can reach 60% in the efficiency of electricity production [22]. Small-scale CHP is commonly used in the UK for heating, and the heat engines include reciprocating internal combustion engine, gas turbine and micro-gas turbine [19]. Heat engines are usually divided into the spark-ignition engine and the compression-ignition engine according to the way of igniting. Table 2.2 shows the properties of three types of engines.

Table 2.2 The properties of three types of engines [19].

Engines	Electrical power range(kW)	Electrical efficiency (%)	Overall efficiency (%)	Advantages	Grade
Internal combustion engine	20-15000	30-45	65-90	Well developed; Low initial investment; Startup easily; Low maintenance cost;	Low and medium temperature
Gas turbine	>900	65-90	65-90	High safety; Low maintenance requirement; Waste heat can be used;	High temperature (steam)
Micro-gas turbine	30-200	75-85	75-85	Compact construction; Low weight; Available to multi-fuel; Easy to control emission;	Low and medium temperature

2.2.2 Heat pump

The heat pump can be categorised into heating-only heat pump, heating and cooling heat pump, integrated heat pump system and heat pump water heaters according to the operational function.

Chua and Chou [23] applied the two-stage heat pump, increasing the overall efficiency by 35%. Park and Jung [24] improved COP about 6% due to the mixture of new refrigerant R170/R290. Chow et al. [25] developed a direct-expansion solar-assisted heat pump, with COP achieving 6.46.

With the attractive advantages of high efficiency and low environmental pollution, heat pumps have been widely applied for heating, cooling and domestic hot water. High grade energy is used by a heat pump as driving energy, while low grade energy such as air, ground water and energy stored in soil can be extracted for terminal users. The heat pump can be divided into air source heat pump (ASHP) system, ground source heat pump system (GSHP) and hybrid heat pump system. Low cost installation is a major benefit of an air source heat pump, which is available for most buildings and can be used for heating and cooling. However, the capacity of an air source heat pump is small, while the temperature of the heat source/sink is fluctuating. In addition, the system needs defrosting and auxiliary equipment for heating at the full load.

Owing to low pollution, high energy efficiency, low maintenance cost, and easy operation, ASHPs have been adopted for building heating/cooling [26]. However, the coil surface of outdoor heat exchanger will be frosted especially in a cold climate, which will decrease COP of ASHP, thus leading to ASHP shutdown. Zhang et al. [26] studied the performance of ASHP used for heating in the coldest region of China, and drew a conclusion that the COP could be acceptable if the indoor and outdoor air temperature difference could be controlled within 41℃. A common solution is reverse cycle of defrosting, the reverse cycle is needed to defrost so that it affects thermal comfort and needs more energy and time for defrosting [27]. Many researchers have made experiments to overcome this problem. Jiang et al. [28] proposed a novel non-frosting air source heat pump system (NASHP) through a change of spray solution, defrosting in a timely and efficient way and enhancing heat transfer. Wang et al. [29] developed a heat pump system, which can avoid frost. Combined with energy storage, it can not only avoid frosting but also supply constant heat in a cycle mode.

The ground source heat pump has the advantages of high efficiency, low energy consumption, low maintenance cost, and energy saving, which is suitable for different heat emitters. However, the installation cost and investment are higher compared with the conventional system. Large space is needed for GSHP system combined with low temperature heating.

The comparison of GSHP technologies from loop type heat source and working depth is shown in Table 2.3 [30]. The loop system of the ground source heat pump can be divided into a vertical loop system and a horizontal loop system [31]. The benefits of ground water heat pump (GWHP) are low cost, and easy to install. In addition, it takes up a small space. However, the maintenance cost is high due to the fouling corrosion in pipes and equipment. The surface water heat pump (SWHP) also has the advantages of low cost because it can save money on digging cost. Moreover, the pumping energy requirements and maintenance cost are low. However, it faces some problems. The coil may be damaged in public lakes and the water temperature is fluctuant due to the weather especially in winter. Regarding ground-coupled heat pumps or closed loop system (GCHP), the pumping energy is also low, while it lacks a stable heat source and is variable in terms of COP under the heating mode.

Table 2.3 GSHP technologies comparison [30]

Category	Loop type	Heat source	Heat source recharge	Typical working depth(m)
GWHP	Open loop	Ground water	Geothermal	6-100
SWHP	Open loop or closed loop	Surface water	Solar irradiation+ geothermal +balance with Atmosphere	0-5
GCHP	Vertical closed loop	soil	Geothermal	6-120
	Horizon closed loop		Solar irradiation+ geothermal	1.5

A district heating heat pump integrated with drinking water network was installed in Milan with the benefits of inhibiting bacteria growth, low risk of water pollution and fouling problems as well as stable water mass flow [32]. Zamfirescu and Dincer [33] developed a high-temperature heat pump with mechanical compression by organic fluids, which used both waste heat and waste matter as the heat source due to a temperature difference of about 50 ℃. The ground source heat pump was widespread in huge energy consuming countries like China due to the high efficiency and low carbon emission [34].

There are some other heat pump technologies to supply heat load such as heat pump combined with solar energy called hybrid heat pump system, which can save 52% of energy monthly for heating compared with traditional space heating [35]. It was obtained that the average COP of a combined solar heat pump system reached 3.7 during the heating season [36]. Dai et al. [37] investigated a solar assisted ground source heat pump system (SAGHPS), which can decrease the duration of soil recovery with the help of solar thermal collector. Ma et al. [38] analysed a solar-ground water heat pump unit and concluded COP of heat pump and overall system increased as the solar fraction rose. Wang et al. [39] developed a doubled-stage coupled heat pumps (DSCHP) for heating, which is suitable for cold regions. Lund and Trygg [40] analysed the feasibility and economical efficiency of using a large scale heat pump in district heating and concluded the heat pump would be widely used for 100% renewable energy supply in the future. Although the heat pump is quite cost-efficient, it is easily influenced by power supply and difficult to combine with other heat sources like solar energy and waste heat from COP [41].

2.2.3 Renewables

Renewable district heating has been introduced all over the world, which mainly focuses on solar energy, geothermal heat and biomass.

The solar energy can be used to supply domestic water like washing at first. With the development of heating technology, soar energy used for space heating attracts public attention. Solar energy has a lot of advantages as follows [42].

i. Solar energy can reduce energy consumption.
ii. Solar energy can use as a stand-by heat source to supply peak heat load.
iii. Solar energy can be used everywhere with some limitations at high latitudes.
iv. The price of solar energy can be predictable and does not depend on the future of other energy prices like coal, oil and natural gas.
v. Solar energy is environmentally friendly.

However, the solar heating is limited by the availability of ground and rooftop space for installing solar collector installation. The high investment cost should be taken into consideration.

Solar energy has been used widely in China, accounting for 70% of the total solar collectors installed in the world, for solar energy resource is abundant, with government subsidies. At first solar energy was used to supply domestic hot water for bath and shower, and then used for heating and cooling due to the development of technology. Zhai, Yang and Wang [43] studied the performance of a solar-powered floor heating system, with the COP of 19.76 and solar fraction of 56%. Zhai and Wang [44] also investigated the cost of solar energy. The payback period for domestic hot water, heating and cooling could be 2 years and 5-8 years respectively. A tri-generation system for heating, cooling, and power generation was proposed by Zhai et al. [45], increasing energy efficiency by 58% and exergy efficiency by 15.2%. It used solar energy as primary energy and the natural gas boiler as auxiliary heat source. Zhao et al. [46] proposed a novel heating system combining solar Kang system with a solar heating system, which avoided auxiliary power and reduced the area of solar collector and energy. The solar fraction of this system can reach 97%. The solar technology for heating also has been developed in other countries. Bauer et al. [47] compared the diffidence of solar district heating between Spanish and Polish with seasonal thermal energy storage. Verma and Murugesan [48] discussed the performance of a solar collector combined with a ground source heat pump system with energy storage, increasing the total COP of the system by 23%. Lizana et al. [49] put forward solar LCE/DH system with biomass boiler as the heat production unit. When solar energy is not enough for heat demand, the biomass boiler will work as an auxiliary heat source for heat supply to fulfil heat demand.

The direct use of geothermal energy for heating can involve much lower quality resources compared with the geothermal generation of power. There is an abundance of geothermal heat, however, the limitation is the city centre location. The direct district heating can only access the heat flow under or beneath the city. The potential is relatively low compared with other renewable heat sources [50]. The development of the geothermal heat pump has been described in Section 2.2.2.

The biomass energy available for district heating can be divided into fuelwood, energy crops, agricultural residues, forestry residues and organic waste [50]. The biomass can be used not only for heat supply but also in power generation and industry [51]. The biomass for district heating in the future is shown in Figure 2.2. Denmark has already used biomass energy for district heating. However, much of this fuel comes from other countries due to the lack of biomass feedstock. On the other hand, Japan and Switzerland benefit from forests and residues also can be used to supply heat. The biomass energy provides a significant potential for district heating supply.

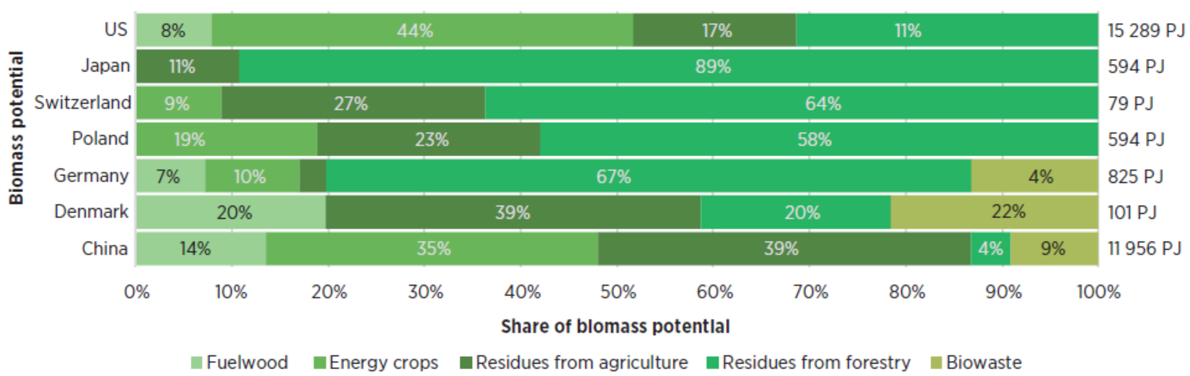


Figure 2.2 Primary Biomass supply potential for district heat production in 2030 [50]

In Sweden, the transition from coal and oil to biomass energy for district heating includes three steps. In the first step from the 1970s to the 1980s due to the oil crises, oil-fired boilers were substituted by co-firing boilers with coal or oil. In 1980, the first batch of oil-fired boilers in CHP were converted to biomass in Växjö [52]. As a result, in the second step was that wood pellets were utilised for heating from the 1980s to avoid energy tax. There were several heating technologies using biomass [53]. A small-scale pellet heating system (SSPHS) became an attractive technology in the 2000s [16]. Pellets are combusted in central heating boilers or stoves [54]. Central heating boilers can be divided into two-unit boilers and integrated boilers. Two-unit boilers consist of a pellet boiler and a standard boiler. The burner is part of the boiler with embedded pellet storage in an integrated boiler. SSPHS has such benefits as lower cost and shorter payback time compared with heat pumps and in lower annual fuel cost compared with district heating. However, the system also should be a hydronic system. To discharge exhaust gas, the chimney is also needed. There is also a need for space to store pellets. The third step involved the investment in new biomass-fired CHP plants. All biomass CHP plants applied traditional steam technology with boilers, condensers, and turbines [52]. In 2009, a biomass-fired CHP with a capacity of 85 MW electricity and 145 MW heat was built in Södertälje. The power-to-heat ratio was 0.59, and the surplus 55 MW of heat was supplied by flue gas condensation. The largest CHP plant was built in Stockholm in 2016, with a capacity of 130 MW electricity and 280 MW heat. The power-to-heat ratio decreased to 0.46 and flue gas condensation recovered 80 MW of heat [52]. In other countries, biomass boilers were always used for heating. Sartor, Quoilin and Dewallef [55] estimated the potential energy saving and environmental performance of CHP biomass plant for district heating in Belgium. Noussan et al. [56] proposed a biomass-fired Organic Ranking Cycle(ORC) unit combined with a heat storage system in Italy, increasing the total efficiency up to 8.6%. Chasapis et al. [57] came up with a hybrid solar-biomass heating system in Greece, which can be connected with conventional heat emitters.

The biomass boiler study is based on a direct system, which provides hot water directly through biomass combustion. The indirect system will use heat exchanger, which can be considered in a thermal storage system. It is assumed that the biomass boiler has sufficient oxygen during combustion, and all the carbon is generated into CO_2 and the water moisture generated can be liquefied to improve the system efficiency. However, the efficiency of biomass boiler can be influenced by the system load factor. Boiler efficiency has a general relationship with system efficiency and its load factor, where the load factor is the ratio of system output to its maximum output capacity. The boiler efficiency will be significantly reduced when the load factor is less than 30% [58].

Biomass heating technology has disadvantages like efficiency, emission, maintenance and more space demand. Another problem is the confusion of the biomass energy for combustion due to the lack of fuel standardisation. However, these disadvantages can be mitigated by advanced technology, optimal design and high quality of fuel (wood pellets), which can lead to low emissions. Biomass energy is extensively utilised in old boilers or heating system is operated and maintained inappropriately, leading to poor indoor quality and health risk. So the quality of such a system and the fuel attract public attention [53].

Overall, the potential for increasing renewable energy use in district heating is significant. There are a lot of barriers during applying the renewable fuel in district heating as Table 2.4 shows

Table 2.4 The barriers for applying renewables in district heating.

Key Barrier	Detail problems
Financing	A large upfront investment cost of district heating projects
Resource availability and cost	Biomass transport logistics and cost; Storage capacity due to inflexible renewables; Uncertain resource availability and environmental impacts;
Constraints imposed by the urban environment and the state of the existing network	Renovation of existing pipework; Appropriate design of new subsystems; Expanding the network;
Policies and regulations	Many of these concepts are new and are not adopted; Permitting procedures related to land use and drilling rights are inefficient

2.3 Heat networks

2.3.1 Insulated pipework

Heat networks are extremely important to heating strategy and have the possibility of helping buildings and industry decarbonise. Insulated pipe network transports energy from source to end-users, and the materials and thermal insulation of pipes impact the pipe reliability and energy loss. Pipes can be categorised into rigid pipes and flexible pipes.

For rigid pipes, the high density polyethylene (HDPE) casing pipe has stagnated in recent years, and researchers have made some progress on thermal oxidation, slow crack growth and rapid crack propagation (RCP) [59]. Smidt and Hansen investigated the oxygen induction time and estimated the life of pipes from 100 to 200 years [60]. Polypropylene variants and one polyethylene grade prepared for blow-moulding applications were studied by

Thörnblom et al. [61]. Nilsson et al. [62] claimed that HDPE grades were not influenced by RCP during temperature changes. Regarding thermal insulation, the lower thermal conductivity materials were studied to reduce heat loss. Polyurethane (PUR) foam was abandoned due to chlorofluorocarbons (CFC). Then, cyclopentane was considered a better alternative. Due to higher vapour pressure and foam containing more gas, HFC-365mfc was much competitive than cyclopentane [63]. However, this gas may cause global warming. The properties of PET (polyethylene terephthalate), aerogel blankets and vacuum insulation panels (VIP) have been studied as alternatives [64, 65].

Flexible pipe systems can be divided into metal and plastic pipes. Flexible pipes have the benefits of reducing installation cost because they can be coiled, free from the need for joints. In addition, straight trenches are not needed. On the other hand, the pressure of the transporting medium is not so high like steel pipes. Pipes are usually made from copper, thin-walled steel, cross-linked polyethylene (PEX) or polybutylene (PB). PEX (Figure 2.3) was used in Sweden in the 1970s with the problem of oxygen diffusion and the benefits of high chemical resistance, simple connecting process and higher temperature resistance to above 95℃ [66].

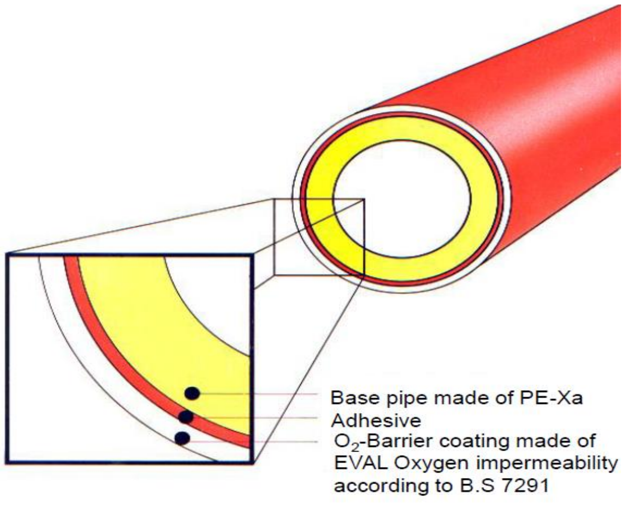


Figure 2.3 The construction of PEX pipe.

To solve this problem, the Swedish GRUDIS concept was proposed in1980s. There was no abnormal damage frequency although the frequency of couplings damage was slightly higher [67]. Compared with PEX systems that need couplings due to unweldable properties, PB is weldable with good temperature resistance [68]. PE foams and mineral wool are widely used for thermal insulation. The advantages, disadvantages and coefficient of thermal conductivity of steel pipe with PU foam, polymer pipe with PU foam and polymer pipe with PEX foam (Figure 2.4) are given in Table 2.5.

Table 2.5 The properties of steel pipe with PU foam, Polymer pipe with PU foam and Polymer pipe with PEX foam [68].

Pipe	Advantages	Disadvantages	Coefficient of thermal conductivity λ ($W/(m \cdot K)$)
Steel pipe (Figure 2.4.a)	Strong material and hard to damage; Larger dimension; Be able to withstand higher flow pressure and temperature;	Joints needed every 6-12 m; Higher cost of excavation on laying pipeline; Only straight lengths possible; Corrosion problems and warning systems and additional polymer galvanized are needed; Specialist welding needed;	0.024
Polymer pipe with PU foam (Figure 2.4.b)	Excellent thermal insulation; No water ingress if the jackets are damaged; No thermal expansion; More flexible than steel; Less joints required due to long coil lengths;	Less flexible than open cell	0.022
Polymer pipe with PEX foam (Figure 2.4.c)	Excellent flexibility; Easy to install and connect; Less joints required due to long coil lengths	Thermal insulation needs improved	0.043

(a) Steel pipe with PU foam (b) Polymer pipe with PU foam (c) Polymer pipe with PEX foam

Figure 2.4 Three typical heating pipes [68].

Pipes can be divided into single pipe (UNO pipe) and twin pipe (DUO pipe) as shown in Figure 2.5. Although the UNO pipe has the strength of no heat between supply and return pipe, the cost of two separate pipes is higher. The cost of DUO pipe is lower than that of two UNO pipes, though there is some heat transfer between supply and return pipe [68].

Figure 2.5 Twin pipe and single pipe [68].

Berge, Adl-Zarrabi, and Hagentoft [69] proposed the conception for hybrid insulation, where vacuum insulation panels are used together with polyurethane foam, as shown in Figure 2.6. It reduces heat transfer between supply and return pipe. The hybrid system is composed of steel pipe and polymer for large district heating systems with high temperature steel mains and the installation of flexible polymer house connections. In addition, the polymer can also be used in branch pipes [70]. Bøhm and Kristjansson [71] studied the triple pipe with two supply lines and one return lines, the heat is supplied by smallest pipe in normal operation while the slighter bigger supply pipe is used for boosting when domestic hot water is needed. It was concluded that triple pipe reduced heat loss by 45% compared with a common pair of single pipes and by 24% compared with circular twin pipes with reduction of investment index by 21% [71].

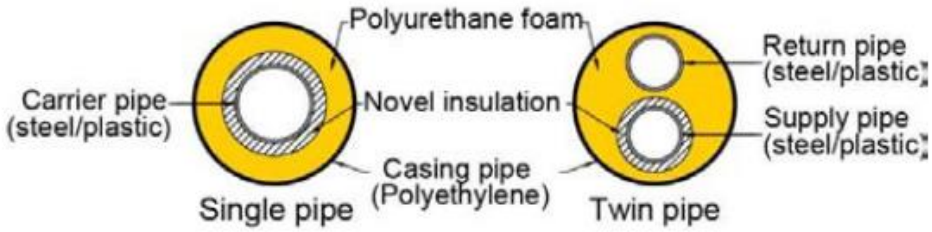


Figure 2.6 Description of the hybrid insulation pipe concepts [69].

Olsen et al. [41] proposed a new DH pipe called AluFlex, which is a multi-layer pipe including PEX and aluminium. Its merits are smooth surface, long lifetime and tightness [41]. AluFlex twin pipe has lower carrier pipe dimension and casing pipe diameter compared with steel twin pipe under the same heat loss [41].

The type of heat network can be divided into radial systems, ring networks and meshed networks according to network size, location of heat source and houses as well as layout of road. The comparison of radial systems, ring networks and meshed networks are shown in Table 2.6.

Table 2.6 The comparison of three types of networks.

Network	Radial systems	Ring networks	Meshed networks
Advantages	Simple network planning; Network type always possible;	Integration of multiple heat sources; Increased supply security;	Optimum supply security; Extension possible;
Disadvantages	Future extension only possible to a small extent;	Only possible with suitable network topology;	High cost, Design mostly for large networks;

Extending pipe lifespan and making a safer network will be considered in the future. It can be concluded from Table 2.7, the lower the supply and return temperature for the same heat load, the smaller pipe size and heat loss. So low supply and return temperature can reduce pipe size, hence lowing capital cost. The LTDH system will be used widely in the future, as discussed in Chapter 2.6.

Table 2.7 Different supply and return temperature wit heat loss and pipe size [72].

Supply/return temperatures (℃)	Heat load (kW)	Pipe size(mm)	Heat loss (kW)	Heat loss saving (%)	Assumption
82/71	450	110	35.2	-	10°C soil temperature;
80/60	450	90	26.4	25%	1300m pipe length; 0.6m installation depth;
80/50	450	75	20.3	43%	2400 operating hours;

2.3.2 Heat Interface Units (HIU)

The district heating system can be divided into direct connection and indirect connection according to the connection type between the DH network and building. In indirection connection, a heat exchanger is necessary, which is used for hydraulic separation between the primary circuit (DH network) and the secondary circuit (space heating).

The most common heat exchanger is plate heat exchanger due to its compact geometry, less surface area and highly efficient heat transfer. Figure 2.7 displays the construction of plate heat exchanger with flow distribution and main dimensions of plates [73]. It includes several plates with gasket, which are pressed together in a frame to make them compact, light and easy to clean [74].

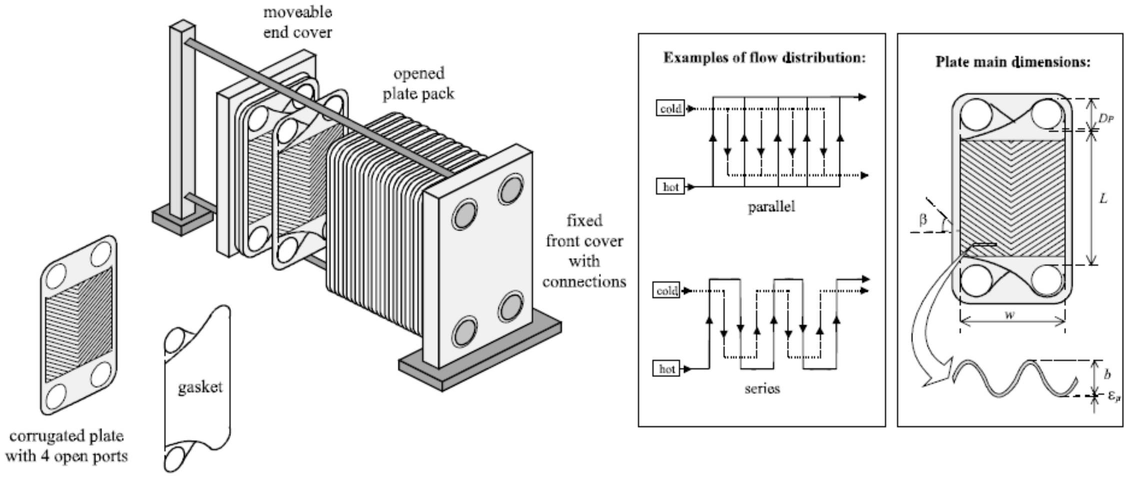


Figure 2.7 The construction of plate heat exchanger with flow distribution and plate main dimensions [62].

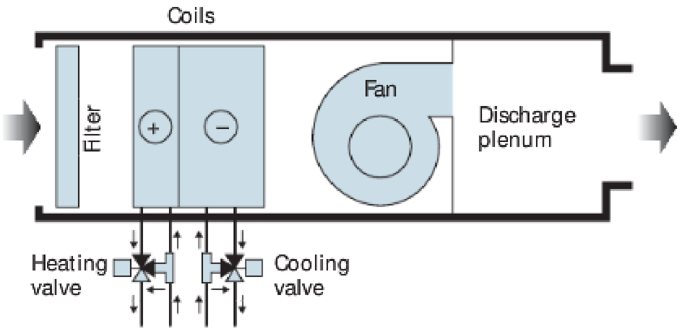
Plate heat exchangers also can be categorised into the brazed plate heat exchanger (Figure 2.8) and plate heat exchanger with gaskets. Brazed heat exchangers are steel units and cannot be dissembled, so it's difficult to internally inspect them. Although the brazed heat exchanger is smaller than the gasketed heat exchanger, the surface area for the brazed heat exchanger is higher. In addition, the brazed heat exchanger has the advantages of reliability and lightweight. However, the problem is low internal water, which may cause temperature problems if the regulating system is not fast enough. It is necessary to use strainers and to be flushed occasionally to avoid fouling [75]. There are several types of heat exchangers like shell and spiral tube heat exchanger, plate fin heat exchanger and spiral heat exchanger [17, 76-78].

Figure 2.8 Brazed plate heat exchanger.

2.3.3 Heat emitters

Heat emitters are used in buildings to supply heat for end-users, and the types of heat emitters are various. Radiators, floor heating and fan coil unit are most used in the heating system as Figure 2.9 shows.

(a) Radiator [79]　　　　(b) Floor heating [80]



(c) Fan coil unit [81]

Figure 2.9 Three types of emitter

Menéndez-Díaz et al. [79] analysed the stoneware-covered emitter, heating up more slowly than the aluminium radiator. Zhou and He [80] studied the performance of low-temperature radiant floor heating system with polyethylene coils and capillary mat under the floor. Wang et al. [82] explored novel floor heating equipment, with lower supply temperature at 30-35 °C. Atienza Márquez et al. [83] used fan-coil and radiant floor as heating units and made a comparison. The advantages of floor heating are lower energy consumption and higher thermal comfort, while the disadvantage is higher thermal mass of the floor which may cause additional energy consumption due to thermal inertia. Compared with floor heating, the temperature adjustment of fan coils is shorter when it is working. However, the floor heating temperature is lower than the water temperature. Myhren and Holmberg [84] studied the pattern and thermal comfort of floor heating and concluded that the installation was costly and difficult especially in renovation of old buildings compared with radiators. However, it improved indoor climate and could lower the temperature diffidence in room due to low temperature compared to radiators. Hasan, Kurnitski and Jokiranta [85] also researched the vertical temperature difference between floor heating and radiator in the water heating system, and concluded that the vertical temperature difference between two methods was small. In addition, floor heating can save living and working space due to installation under the surface using ducts without production of noise [86].

2.4 Thermal storage

Thermal energy storage (TES) is utilised to store energy and then later used for heating and cooling. Thermal storage has been widely applied to integrate more renewable energy, balance fluctuating resources and improve energy efficiency [87]. Figure 2.10 shows the charging process from heat source to TES and the discharging process from TES to users [88].

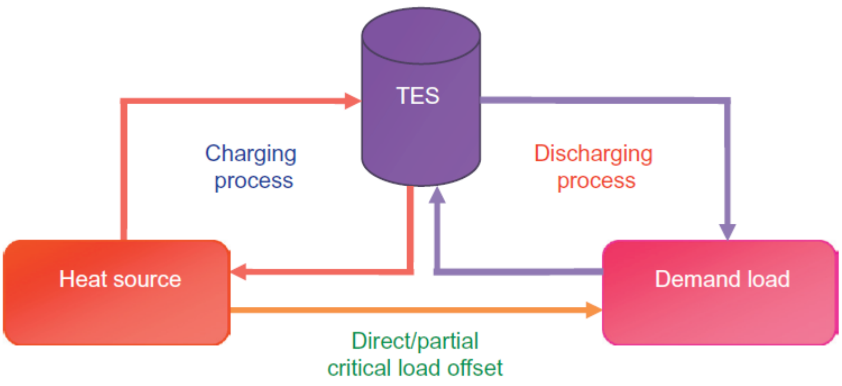


Figure 2.10 The Operating principle of thermal energy storage [88]

Thermal energy storage can be classified into sensible energy storage, latent energy storage and chemical storage as Figure 2.11 shows. Compared with latent heat storage and chemical storage technology, sensible heat storage technology is the simplest method.

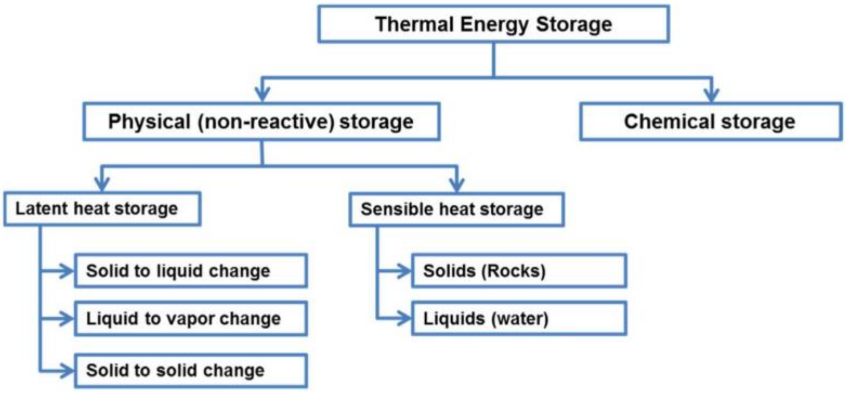


Figure 2.11 Thermal energy storage methods.

The materials of sensible heat storage can be divided into liquid and solid materials, where liquid materials are commonly used. Water is the best choice due to high heat capacity as Figure 2.12 shows.

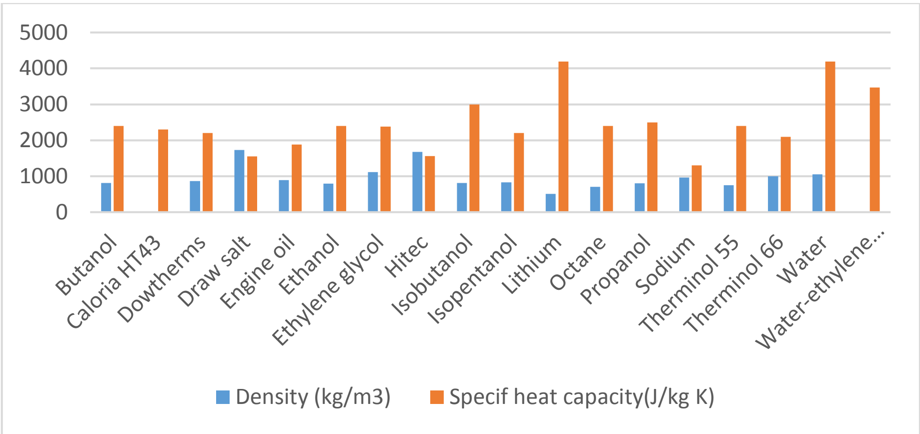


Figure 2.12 Properties of liquid sensible heat storage materials [80].

Table 2.8 shows the capacity, power, efficiency and cost of storage technologies. Although chemical reaction has high capacity and efficiency, the cost of chemical reaction is higher than others. For seasonal energy storage, hot water is chosen to store energy. In general, water is the best heat storage medium because it is inexpensive and has no risk deriving from the use of toxic materials, with high specific heat [87].

Table 2.8 The properties of different storage technology [87].

Storage Technologies	Capacity (kWh/t)	Power (MW)	Efficiency	Storage Time	Cost (cent/kWh)
Hot water	10-50	0.001-10	50-90%	day-year	0.01
PCM	50-150	0.001-1	75-90%	hour-week	1-5
Chemical Reaction	120-250	0.01-1	100%	hour-week	1.8-4

2.5 Trends of district heating technologies

Table 2.9 shows the transformation of the district heating system from 1980 to today. The heat carrier for district heating was steam until 1930, and the first generation used pipes in concrete ducts, steam traps and compensators. This technology adopted coal steam boilers or some CHP plants to replace individual boilers so that the accidents from boiler explosions were fewer than before and it increased thermal comfort.

Table 2.9 Summary of the first three Generation [89].

	1st Generation	2nd Generation	3rd Generation
Time	From 1880 to 1930	From 1930 to 1980	From 1980 to 2020
Property of pipe	In situ insulated steel pipes	In situ insulated steel pipes	Pre-insulated steel pipes
Transport fluid	steam	Pressurized hot water mostly over 100 °C	Pressurized hot water often below 100 °C
Circulate system	Steam pressure	Central pumps	Central pumps
Substation heat Exchanger	NO	Tube-and-shell heat exchangers	With or without plate heat exchanger
Buildings	Apartment and service sector building in the city	Apartment and service sector building in the city 200-300kWh/m^2	Apartment and service sector building in the city (and some single-family houses) 100-200 kWh/m^2
Heat emitter	High-temperature radiators (+90 °C) using steam or water	High-temperature radiators (90 °C) using district heating water in direct or indirect system	Medium-temperature radiators (70 °C) using district heating water in direct or indirect system
Heat production	Steam boilers	CHP and heat-only boiler	Larger-scale CHP

The risk of steam explosions and heat loss, however, increased substantially. The second generation used pressurised hot water as the transport fluid, and the temperature of hot water mostly was above 100℃. Concrete ducts, tube-and-shell heat exchangers and valves are essential components in pressurised hot water pipes. Better comfort and fuel saving can be achieved by using coal and oil based CHP or some heat-only boilers, but the disadvantage of this system is the lack of heat demand control. From 1970s Scandinavia developed the third generation which also used pressurised hot water. Nevertheless, the temperature of water was below 100 °C so that it was featured by prefabricated and buried steel pipes as well as plate heat exchanger. It was used widely due to the security of supply and policy, and there were two oil crises so that a new technology with energy efficiency was needed to overcome dilemmas [16, 89, 90]. A large-scale CHP, biomass and waste boiler were used to produce heat as well as fossil fuel boilers, which was combined the existing heating facilities with a mixture of renewable energy and conventional fuels at a reasonable cost [91].

Nowadays, the focus of district heating shifts to a more sustainable system with less carbon emission [92]. The LTDH system is introduced, which will be described in detail in Chapter 2.6. Figure 2.13 shows community energy diagram for heating, cooling and electricity by oil, biomass, natural gas, coal and renewable energy source with thermal storage, which is applied in the next decades.

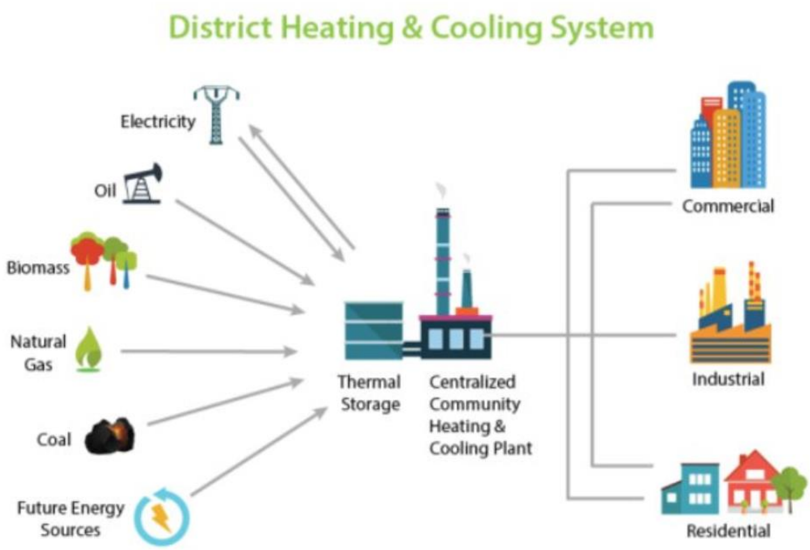


Figure 2.13 The diagram of district heating and cooling system [92].

District heating has evolved considerably over the last couple of years [92]. Some respective countries are chosen to describe in terms of latitude and climate, namely China, Russia, America, EU countries.

The district heating development in China increased significantly by 10-15% a year which has been the highest growth rate since 1998. The total building area served was 5.1 billion square meters in 2012 as Figure 2.14 shows, it is expected to rise to about 7 billion square meters in 2020 [93]. This expansion can primarily results from pollution and population. China has the second largest installed capacity of CHP plants across the world. CHP supplies 30% of heat demand, and the figure is expected to double in 2020 [92].

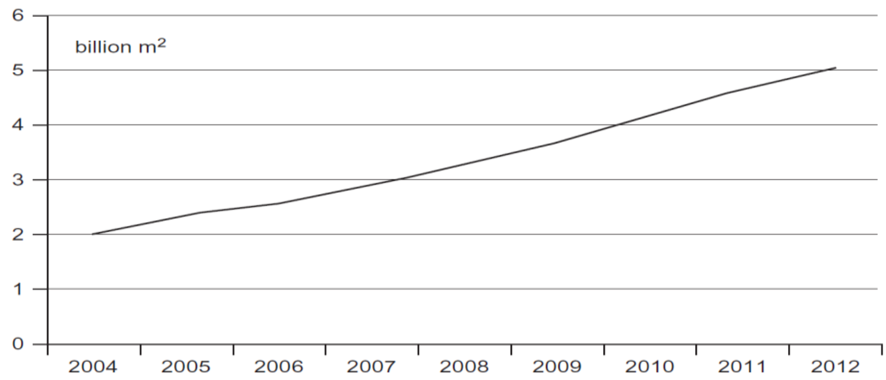


Figure 2.14 The expansion of DH with steam and water as carriers from 2004 to 2012 [92].

Russia has the largest district heating system, containing 500 large CHPs and 6500 large-scale boilers and 20000 km networks. 30% of heat is supplied by CHP and 45% by heat boilers, while the remaining is produced by industrial and other heat sources [94, 95]. District heating system supplies 70-80% of heat demand, serving 70% of population in the entire country [95].

America had 600 district heating schemes with the installed thermal capacity of 16.6 GW. The number of these schemes located in the city centre was 106, of which 55 had CHP-DH networks while the remaining scheme were found in campuses and hospitals. Nowadays, the USA has an installed capacity of about 82 GW, of which 80% is related to the industrial sector. The district heating in USA unlike EU countries, is mostly supplied to commercial buildings instead of residential buildings [92].

The district heating in Europe is comparatively well evolved. East Europe and Scandinavia have a long association with district heating, while central Europe has developed the DH with sustainable heating sources [92]. Denmark developed the DH technology and became the world leader due to the oil crisis in 1970. Denmark had 285 decentralized CHP plants, covering 46% of heating demand of the entire country, 50% of district heating was supplied by large scale CHPs and 20% was supplied by small scale CHPs [96]. There were 16 large decentralized CHP plants and 130 plants with a backup boiler [97]. District heating networks in Finland started in the early 1950s [98]. The total demand for heating was about 37 TWh, 36.5% of which was CHP and 20% was renewable energy [92]. Nowadays, district heating in German coveres 14% of space heating demands, which is expected to reach 25% of electricity generated via CHPs [92].

The percentage of applying DH systems in EU countries is shown in Figure 2.15. Iceland, Latvia, Denmark, many Eastern European countries turn to district heating, but district heating in the UK is not popular.

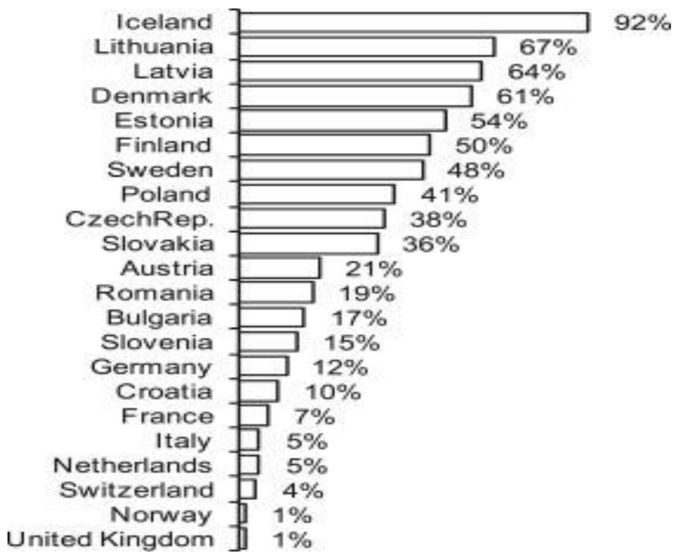


Figure 2.15 EU countries participation in DH systems [91].

For the UK, there were several buildings needing to be replaced with a high population growth rate from 1950 and 1975. The district heating system was developed to solve the heating problems and reduce pollution. The heat was supplied by central boilers which used fuel oil due to the low price until 1970. Then, the heat source was converted to natural gas. District heating was being developed from 1960 to 1970. Because the heat meters were not installed, the heat was wasted. Moreover, users were not satisfied with the thermal comfort. As a result, this system was replaced by individual gas-fired boilers. UK has achieved natural gas from the North Sea since 1970, and the individual natural gas heating became the main choice. As a result, the gas boilers contribute to 85% of domestic heating now, while district heating accounts for 2% [99]. There were some developments in district heating. For example, National Heat Boarder made a report about large scale district heating in 1979. In 1986, Southampton began to use CHP plant alongside absorption chillers and backup vapour compression machines with the thermal heat for heat supply and cooking in the city [100]. In 1987, Sheffield established one of the oldest district heating networks in the UK, which was continuously expanded until the present. The main fuel source was a waste incinerator, covering 2800 homes and 140 public and private buildings. The Nottingham district heating scheme originated from 1989 to the present, which covered 5000 domestic consumers. In 1992, the district heating and cooling system was established in London. In Scotland, the Aberdeen Heat and Power scheme was set up on a small scale, which is continuing to expand nowadays. The University of Warwick installed gas CHP plant in 2001 and extended the network from 16 km to 19 km in 2014, which supplied 60% of its electricity, heating and cooling demand. In 2010, the Queen Elizabeth Olympic Park District Energy Scheme was developed, which was one of the largest combined cooling, heating and power facilities in the UK. The heating demand was supplied by a 3.5 MW woodchip biomass boiler, three 3.3 MW gas-engines and one 80MW backup hot water boiler. The cooking demand was covered by 4MW absorption chillers. In 2016, Gateshead Council together with Parsons Brinckerhoff used 2

MW gas CHP engines to supply 35 GWh of electricity per year. The University of St Andrews improved district heating design and lowered heat loss in 2017, which was one of the largest biomass and district heating schemes in Scotland [100].

2.6 Low Temperature District Heating technology

2.6.1 Concept of Low Temperature District Heating

The trend throughout the three generations has been to lower energy consumption, temperatures and carbon emissions, so the Low Temperature District Heating (hereafter referred to as fourth generation district heating 4GDH) has been proposed. The aim of heating in Denmark is to fulfil heat supply by completely relying on renewable energy in 2050. To achieve this objective, in 2007, the concept of LTDH was proposed in Denmark in the project of development and demonstration of low-energy district heating for low-energy buildings. Then, the Energy Technology Development and Demonstration Programme (EUDP) applied LTDH in Lystrup in 2008. In 2010, the next step of EUDP was to improve the concept of LTDH and renovate old buildings for LTDH [101].

Unlike the above three generations, the 4GDH integrated fluctuating and intermittent renewable energy source with a conventional heating source to supply heat for users [89]. The supply and return temperature of this system are usually set at about 50/20 ℃, which decreases significantly heat loss of networks compared with the 3GDH due to the temperature difference. There is a much higher potential for using renewable energy. Pre-insulated flexible (possible twin) pipes are widely used in LTDH system, with the installation of floor heating and low temperature radiators in the indirect system. Heat production derives from CHP together with renewable energy (e.g. biomass boiler, solar energy and heat pump). The 4th Generation District Heating entails heat storage to overcome the fluctuation and intermittency of renewable energy such as solar energy.

The LTDH has some additional benefits because of the low supply and return temperatures in distributions, which can be summarised as follows [89]:

i. higher power-to-heat rate in steam CHP rate
ii. higher coefficient of performance in heat pump
iii. higher heat recovery from gas condensation
iv. higher utilisation of industrial and geothermal heat sources
v. higher conversion efficiency in central solar collector
vi. higher capacity in thermal energy storage if they can be charged to a temperature above the ordinary supply temperature

A comparison among the four generations of district heating technologies from temperature level, energy efficiency and heat production is shown in Figure 2.16.

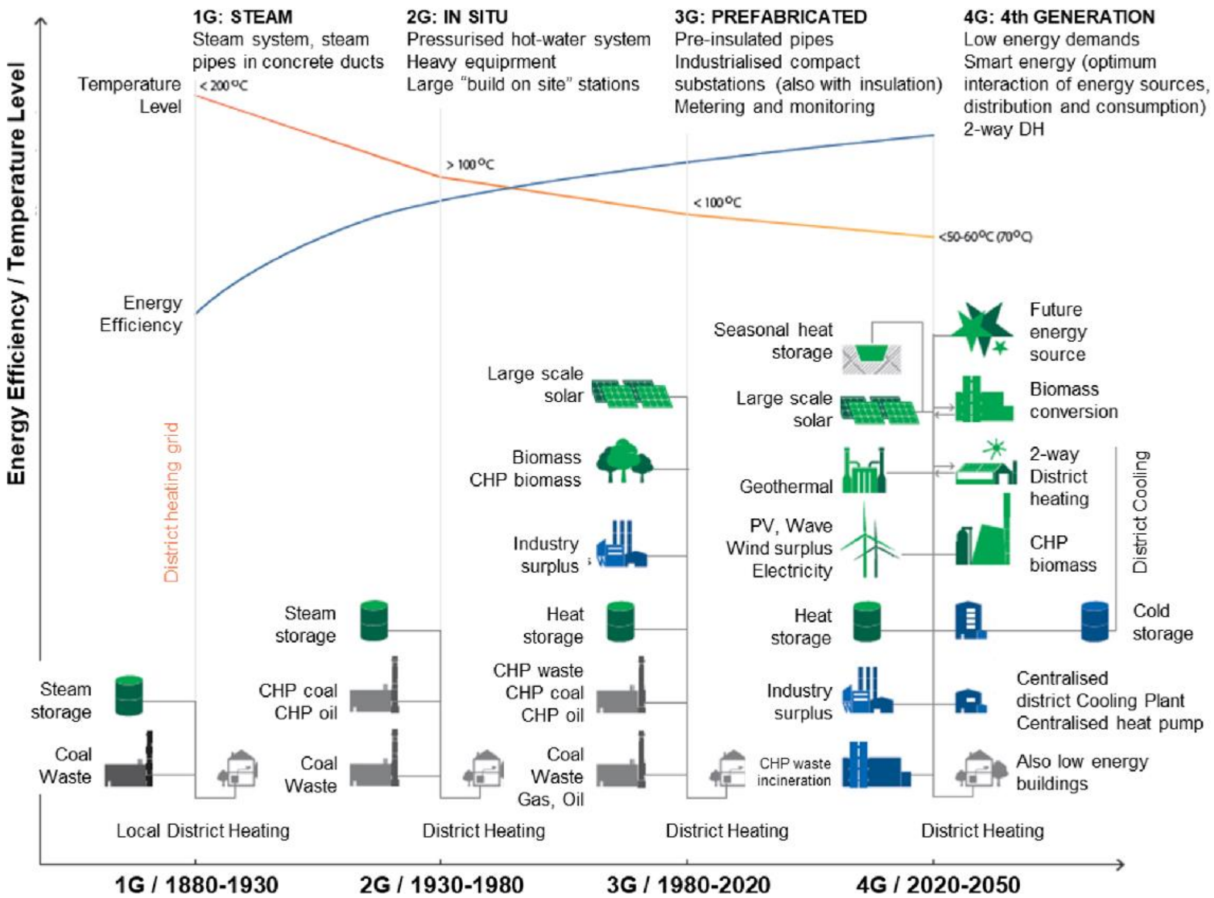


Figure 2.16 The comparison between 4th generation district heating and the previous three generations [89]

Dalla Rosa and Christensen [102] made a cost comparison between LTDH and ground source heat pump for heating energy-efficient buildings and concluded LTDH relying on renewable energy was better from environmental and economic point of view. The distributed heat loss in LTDH was halved and the pipe size was reduced, which contributed to lowering the investment cost. Olsen et al. [41] calculated social costs including maintenance and operation cost, investment and re-investment cost, and taxes for LTDH, ground coil heat pump and air-to-water heat pump in a 30-year period. The results show LTDH is more competitive than the heat pump.

Østergaard and Lund [103] introduced an absorption heat pump using low temperature geothermal energy for heating in Frederikshavn, Denmark. Zvingilaite et al. [104] studied the Low Temperature District Heating with the supply temperature of 40 °C, using a micro-heat pump for domestic hot water, which is competitive than the electrical heater due to the energy price and future socioeconomic costs. In China, there is substantial industrial waste heat, which brings a huge potential for LTDH. Fang et al. [105] proposed different systems according to terminals to make return temperature below 30 °C or even 20 ℃.

2.6.2 Space heating and domestic hot water preparation with LTDH

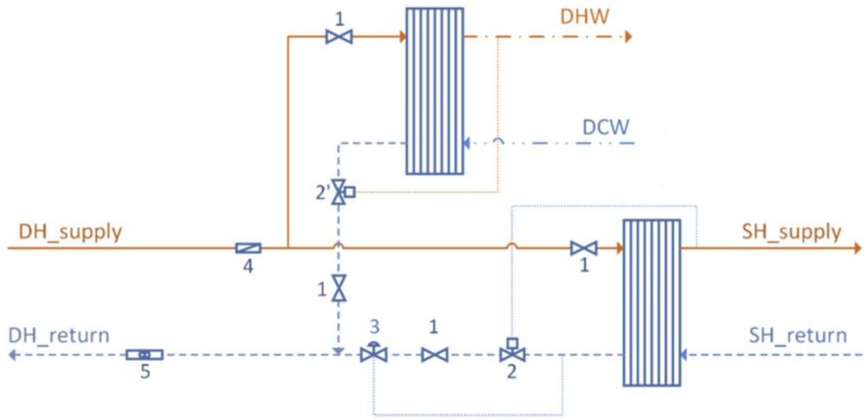
Owning to the merits of LTDH system, it has attracted public attention. However, there are some obstacles to realise the LTDH system, which supplies heat for domestic hot water and space heating.

One of the obstacles is Legionella in domestic hot water. It is generally believed that Legionella may increase significantly temperatures from approximately 20 °C to 50 °C Table 2.10 shows the temperature standards for designing hot water system in different countries.

Table 2.10 Examples of temperature used for design hot water system [27]

Country	Denmark	Finland	Korea	Russia	United Kingdom	Poland	Germany
Hot water (℃)	<60	55	55	50	65	55	55

The supply temperature of LTDH is about 50-55 ℃, and many solutions have been developed to solve this problem. According to German standard DVGW, there will be no Legionella risk even the temperature below 50 ℃, if the overall volume of the DHW system excluding heat exchanger is below 3L [106]. Decentralized substations installed in each flat (Figure 2.17) were proposed by Yang, Li and Svendsen to eliminate Legionella risk with a small volume of domestic hot water, while lower return temperature was restricted by high-temperature bypass flow [15].



1. Ball valve 2. Thermostatic valve 2'.Thermostatic valve with bypass function 3.Differential pressure controller 4.Strainer 5.Energy meter

Figure 2.17 Connection of decentralized substation unit [15]

Paulsen et al. used the original substation with 200L thermal store for domestic water and space heating [107]. The tank was used as a buffer tank to supply DH water as shown in Figure 2.18. There is no tank for domestic hot water, which can be heated in a heat exchanger if necessary. Although the Legionella can be avoided in this system, the large heat loss may occur. Another method was the use of AOT (Advanced Oxidation Technologies), which was not widely applied due to the significant investment and running cost.

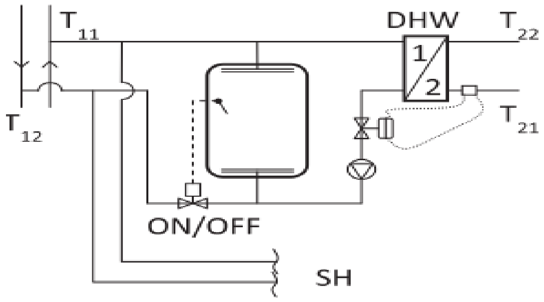


Figure 2.18 Principle of low-temperature DH substation with buffer tank for DH water [107]

For space heating, it is an opportunity to lower supply temperature at about 50 ℃ in new building due to the insulated building envelope, floor heating and low-temperature radiators [108]. However, there are substantial old buildings which need more heating demand. Building envelopes are improved and original space heating systems are replaced by the low-temperature system. It is a faster and cheaper method to change the space heating system, however, it cannot save energy. Insulation of roof and walls can improve the building envelope, but the cost is not accepted by householders [104]. Another typical method is to replace windows, which is appreciated due to the long lifetime, small investment and significant energy saving [108]. In addition, radiators, which are over-dimensioned design, are usually designed for more heat. It is beneficial to lower supply temperature and does not affect heat comfort for end-users.

2.7 Policies and outlook

The development of district heating is affected by the aspects of energy, climatic and local conditions, environment and heat demand. The policies of each country are different due to the impacts of these aspects. Meanwhile, the differences in problems and outlook faced exist in each country according to the aim and objective.

The objective of Chinese government was to increase the sustainable energy share to 15% by 2020 and decrease the carbon emission by 40-45% compared to the values in 2005 [109]. In 2015, 'Intended Nationally Determined Contribution' (INDC) document announced that the goal was to peak its emission around 2030 and to lower the carbon intensity of GDP by 60-65% of the level in 2005 by 2030 [110]. Based on the previous research, the district heating could reduce 60% of heating consumption and 15% of cost at the current level [93]. The best choice to reduce the carbon emission and energy consumption and cover the heat demand is district heating [93]. However, the challenge for China is greenhouse gas emission and the competitiveness of district heating. 70% of people in China live in urban centres and face harmful air according to World Health Organization (WTO) [92]. This is why district heating technology is widely applied in China. A lack of control and monitoring equipment in suppliers and end-users is the main reason to cause the more waste heat, low response time and less flexibility of district heating technology compared to EU countries.

The main drivers for the largest district heating in Russia were the policy before 1990 and the climate [111]. Then, the way of heating was a boiler, which was based on the district heating grid due to plentiful fuels. The heating system was inefficient, with a high carbon emission. The aim of Russia in the energy efficiency strategy is to reduce the heat loss from 20% to 10% by 2030 [112]. At the moment the biggest challenge will be that networks are too old. 60% of district heating networks were needed to be repaired and some pipelines were more than 50 years old [113]. The heat loss of some pipelines can reach 50% compared to the 5-10% heat loss in EU [113]. Another challenge is a lack of market competitive in networks, and most networks are state-owned. The reform from heat source, end-users and networks is required and government financial support is also needed [114].

US federal government set an aim of raising CHPs capacity by 50% to 122GW by 2020 [115]. 75% of the states in the US have finical incentives to develop CHP schemes with a sustainable system. The US promotes DHC to the roadmap of district heating. At the moment the main challenge of district heating is high investment and low payback rate. The government will introduce heat tariffs, population awareness and specific DH policies in the future [92].

In 2009 the European Parliament presented long term targets in different areas of energy policy [116]. Firstly, the greenhouse gas would reduce 20% by 2020 and 80-90% by 2030 compared to 1990. The next one is that the renewable technology would generate 20% of final energy demand by 2020 and 30% by 2030. Finally, the energy efficiency target is to decrease 20% of energy consumption by 2020.

Each EU member has respective policy and outlook to achieve the target of EU policies. The national regulation on promoting DH technology in Denmark was presented after the 1970 oil crisis. The aim of Denmark is to be independent of fossil fuels by 2050 [117]. It can be achieved by the following steps. Firstly, the large heat pump installed in the district heating system adopts the excess power produced from the wine turbine. The next one is that is the large-scale coal-fired power plants are converted to biomass and a smaller part of DH is covered by CHP and renewables. Finally, the old building needing to be retrofitted with high insulation and more low energy should be constructed [117]. The biggest challenge is a lack of competitiveness in DH because most of the generation plants are owned by local authorities and prices are highly regulated. The Finnish government had the same objective of fossil-fuel free like Denmark. Most of centralized heat production plants based on fossil fuel were transformed to use biomass and wood as a heat source [98]. The challenge is security of operation by wood and biogas as a heat source and the roadmap of heating in the future with renewables and heat pump is not defined [118]. Germany government set the aim to reduce 80% of greenhouse gas by 2020 compared to the levels of 1990 and fulfil 100% of renewable energy system by 2050 like EU directive [119]. The capacity of CHPs increases by 100% by 2020 according to the CHP and DH laws [120]. Unlike other EU members, German promoted decentralized small-scale heating sources to a regional grid. German policies advocated excess electricity-driven heat pump. The challenge is high investment and operation control. The policy of the Polish government was to improve the efficiency of DH heating, decrease greenhouse gas emission and promote the CHPs instead of large scale boilers [121]. In the future, the heat sources will transform from traditional fuels to renewable energy like biomass, waste industrial heat and solar energy. The heating end needs upgrading like installing meters and retrofitting buildings. The problem encountered is keeping a completive price.

Under the Climate Change Act 2008, the UK government has committed itself to reducing greenhouse gas emission by more than 80% by 2050 and achieving net zero emission across the country [122]. The percentage of district heating used by households in the UK is very small. It is too hard for the government to achieve the objectives. As a result, the government announced other policies related to energy efficiency, low carbon heat and low carbon gas. The polices can be divided into the following aspects:

i) Incentives. The UK government has launched an £ 860 million Renewable Heat Incentive (RHI), making payments to households available from October 2012. The Department of Energy and Climate Change (DECC) says the scheme is expected to increase green capital investment by £4.5 billion by 2020, and increase the number of industrial, commercial and public sector renewable heat installations sevenfold by 2020 [123-126]. The Renewable Heat Incentive (RHI) is a government financial incentive to promote the use of renewable heat. By switching to heating systems that use renewable energy, it can help the UK reduce its carbon emissions [127]. The Renewable Heat Incentive has two schemes - Domestic and Non-Domestic [123]. They have separate tariffs, joining conditions, rules and application processes.

ii) Obligations. The Renewables Obligation (RO) came into effect to support mechanisms for large-scale renewable electricity projects. As a result, the proportion of electricity from renewable sources was increasing [128]. Renewable Transport Fuel Obligation was published in 2012 to support the production of biogases which can be injected into the gas grid [129]. In 2013, the government provided the Energy Company Obligation (ECO) scheme, which drove the improvement of energy efficiency in fuel deficient households [130].

iii) Regulation. The government introduced technology regulation to ask the boilers to be installed in the new buildings or existing boilers should be replaced by the condensing boiler with 15-30% higher efficiency [131]. Then, the Boiler Plus regulation was introduced in 2018 to strengthen boiler efficiency, which requires all the boilers have a minimum efficiency of 92% [132]. The Part L of Building Regulation was established to specify the minimum energy performance requirement for new and existing buildings, which decreased the heat demand of buildings [133]. In 2015, the minimum energy efficiency standard was provided to increase energy efficiency, which prohibited the lands from letting properties if the rate was below EPC (Energy Performance Certificate) B and C [134].

iv) Taxation and levels. Both electricity and gas are subjected to taxation and a reduction of VAT [135]. Most incentives and levels are only applied to electricity, including Renewable Obligation, capacity payments, and the carbon price floor. Levels for smart meter payments, the Energy Company Obligation and the Warm Home Discount are shared between gas and electricity [135].

In addition, there were some other projects to support the aim and objectives. Heat Networks Deliver Unit (HNDU) was launched in 2013 to provide funding and specialist guidance for local authorities to develop the heat networks [136]. In 2018, the UK government strengthened the funding to propose Heat Networks Investment project (HNIP), which invested £ 320m of capital funding in heat network project through loans and grants. It can help to develop the heat networks market [137].

2.8 Summary

An introduction to DH systems was highlighted in this chapter, providing main concepts and a detailed description of this technology. The analysis covered the ability of DH to make use of fossil fuels and renewable sources. In addition, the development of heat networks for the DH were described including insulated pipework, HIU and heat emitters. The comparison of different types of thermal storage were made from heat capacity and cost. Water is commonly used in DH due to high heat capacity, low cost and non-toxicity. All these developments illustrated the potentials in the transition towards a low carbon economy.

A detailed overview was provided about the actual heating sector from 1st generation to 3rd generation in the world and the relative market. It was assessed the DH penetration in Russia, China, USA and the EU energy market. To this extent, the case of UK among the others, was analysed more in details. A detailed description was provided about the current DH share. In fact, the DH technology is not popular in UK.

This review identified the background necessary to introduce the concept of LTDH, which is the core of the present research and links DH technology to the UK actual and future context as this was one of the main drivers of the research work of this investigation.

The policies in the world to release energy saving and low carbon emission were summarized, illustrating the role of DH systems in the transition towards a low carbon economy and 100% renewable energy system. The UK is recognised within the EU context as an emerging country for DH. District heating technology development and it became central in the national political agenda as one of the key technologies to decarbonise the UK heating sector. Finally, it was presented an outline of the barriers for the UK energy sector that limited in the past the deployment, competitiveness and the reliability of DH technology.

3 Energy demand assessment of the site: Case of Creative Energy Homes (CEHs)

3.1 Introduction

This chapter describes the use of dynamic thermal models to simulate the space heating and domestic hot water heating consumption and heating load for CEHs.

In this study, EnergyPlus is a freely available dynamic thermal modelling tool which has undergone several revisions and the current version 9.1.0 was released in March 2019. The input data for EnergyPlus simulations is contained in a text file called the Input Data File (IDF). This enables the user to change sections of the input file and control these changes using a text editor or a third party such as IDF editor.

SketchUp Euclid is a free and open-source extension for SketchUp that makes it easy to create and modify the geometry inputs for building energy models, for which it uses the EnergyPlus simulation engine and provides a user-friendly graphical user interface. In this study, SketchUp version 2018 (March 2018 release) and Euclid version 0.9.3 (April 2017 release) were used to input the building geometries. The model created in SketchUp were then converted to the EnergyPlus IDF files, which were modified further using a text editor and the EnergyPlus IDF Editor to construct the final EnergyPlus model and run simulations

The chapter starts with the description of modelling house construction system, heating and cooling system (section 3.2). Then in section 3.3, it describes the reason for using EnergyPlus and the Energyplus method. In section 3.4, the procedure for establishing models and the parameters like the thermal transmittance (U-value), heating and indoor temperature setpoint, internal heat has been defined. In section 3.5, the results of creative house are obtained, which then are analysed. Finally, 3.6 presents a summary of this chapter.

3.2 Site description

3.2.1 Background

The Creative Energy Homes project located at the Green Close on the University of Nottingham campus. Figure 3.1 shows the Creative Energy Homes site. There are 7 buildings in total that will utilise a range of renewable energy and micro-generation technologies, including biomass boilers, solar-photovoltaics, micro-wind, air source and ground source heat pump, solar thermal system.

1—Mark Group House; 2—Nottingham House; 3—BASF House; 4—EON House; 5—Tarmac House (Code 6); 6— Tarmac House (Code 4); 7—David Wilson House

Figure 3.1 The Creative Energy Homes site

The buildings on the site are named after the construction company that built them. Table 3.1 shows the name of each buildings of Figure 3.1.

Table 3.1 Year construction of buildings

Building Number	Building name	Year of construction
1	Mark Group	2010
2	Nottingham	2010
3	BASF	2008
4	EON	2011
5	Tarmac (Code 6)	2010
6	Tarmac (Code 4)	2010
7	David Wilson	1999

Two Tarmac houses are separately Code for Sustainable Homes (CfSH) level 4 and level 6. The Code for Sustainable Homes is an environmental assessment method for rating and certifying the performance of new homes, which covers energy, CO_2 emissions, water, surface water run-off, materials, waste, health and well-being, management, pollution and ecology [138, 139]. The determination of CfSH can be defined by the CfSH credits. Table 3.2 shows the construction example for CfSH credits.

Table 3.2 The construction CfSH credits for Tarmac House code 4 and 6.

Element	Tarmac House code 4 CfSH credits	Tarmac House code 6 CfSH credits
External walls	3	3
Floor	3	1.75
Internal walls	2.45	2.45
Roof	3	3
Windows	2	3
Total credits	13.45	13.2

3.2.2 Construction and technology descriptions of the dwellings

The building envelope acts as a climate moderator, which provides a balance between the heat gains and heat loss required to maintain a comfortable interior temperature.

Many of the seven houses are constructed by the Modern Methods of Construction (MMC), which was introduced to deliver more houses of better quality at a faster rate in the last few years, most of which use lightweight materials like timber, lightweight steel frame[140]. As a result, the materials can produce highly insulated buildings with low-infiltration rates through the envelope [140, 141].

Energy consumption on buildings is space heating, domestic hot water, cooking, lighting and appliances. Due to the weather of the UK, the cooling energy consumption is much less than heating [6]. The heating including space heating and domestic hot water accounted for 81% of total energy consumption. As a result, the heating and cooling technology is a significant factor to achieve energy efficient buildings.

In addition, the CEHs installed many renewable energy and low carbon technologies to enhance the energy performance of the houses, some of them are listed in the Table 3.3.

Table 3.3 Heating and cooling technology in CEHs.

House	Heating and cooling technology
BASF	Earth to air heat exchangers Biomass boiler Solar thermal system
David Wilson	Ground source heat pump Solar thermal system Gas boiler
Eon	Mechanical ventilation with heat recovery Electric and gas-fired heat pump
Mark group	Ground source heat pump SUNWARM Solar hot water system
Nottingham	Passive downdraught evaporative cooling Solar hot water system
Tarmac	Biomass boiler Mechanical ventilation with heat recovery Solar hot water

3.3 Methodology

The real data for CEHs will require full access to the dwellings and observe ethical issues of residents' data. The work is to conduct a validation analysis of the simulation data using the LTDH network of the Creative Energy Homes. The CEHs performance can be simulated by software. The building performance simulation (BPS) software available today and to have confidence in the predictions of whole-building energy models, it is necessary to have a thorough understanding of various features, specific capabilities [142]. All of the University of Nottingham's CEHs (Mark Group, BASF, Nottingham, Tarmac, Eon, and David Wilson Houses) energy performance needed to simulate and then the building performance for the CEHs can be obtained. Today, the software EnergyPlus and IES are widely used as BPS tools [142].

3.3.1 Justification

EnergyPlus is used to model the heating demand and heating load on-site. EnergyPlus, developed by the U.S. Department of Energy, is a whole building energy simulation program based on a modular structure that has shown a continuous enhancement in the possibility of adding validated new models [143]. Integrated Environmental Solutions Virtual Environment (IES VE) is another comprehensive whole-building simulation tool that provides design professionals with a single software environment for a detailed assessment and optimisation of building and system designs [144, 145]. For instance [146], a survey was conducted that 108 modelers from engineering and architectural companies involved in a series of national and international projects at stage of the design process, and reported that 80% of respondents choose IES as their energy analysis simulation tool. Although EnergyPlus can be used as a stand-alone tool, the main obstacle to the widespread adoption of the technology by practitioners is the lack of a comprehensive graphical user interface for the rapid development of building geometry. This situation has changed over the past decade, and packages have been developed that either use EnergyPlus as their main simulation tool (such as DesignBuilder, Sefaira, and OpenStudio) or have plugins for integrating with them

(Eg SketchUp, Revit). As a result, EnergyPlus has become easier for architects and other professionals to use [142].

EnegyPlus software offers some advantages to model the energy demand of the houses which include [142-145, 147]:

- Widely used in the market for calculating energy consumption.
- Can take into account of a wide range of function such as radiation of the building and sky, natural lighting
- Capabilities of analysis solar, climate, carbon, fossil, regulations, global compliance and value/cost/environmental impact.
- Easy visualisation and communication of results.
- Clear diagrammatical and graphical outputs

3.3.2 EnergyPlus modelling method

The following steps were used to generate and analyse the CEHs:

i. Model built using Sketchup Pro 2017 by importing CAD as-built layouts
ii. Sketchup plugin extensions for Sketchup is used to define rooms, windows, and door.
iii. The model is imported to EnergyPlus through the extension
iv. Building features can be then edited
v. Room occupancy
vi. Location (Including CIBSE weather data)
vii. Build material properties (e.g. U-value, transmittance, materials)
viii. Water consumption
ix. Energy system types and efficiencies
x. Heat gains (people, electrical equipment)
xi. Airflow
xii. The temperature setpoint for each room
xiii. The time for people stayed in home and working time for electrical equipment

xiv. Define the variable output parameter (hourly heating load, monthly heating demand, outdoor temperature)

xv. Data is then exported to Excel where it can be manipulated for further analysis (see Results)

3.4 EnergyPlus modelling of the buildings

The site features have been described in section 3.1, the heat demand for the site can be calculated from the following steps. Firstly, models for seven houses are built using SketchUp Pro 2018 by importing CAD as built layout.

Then, EnergyPlus plugin extensions for SketchUp is used to define rooms, windows and doors. Figure 3.2 shows the model for Nottingham house [148]. The simplification of the model ignores the interior windows and interior doors.

The model for all the other buildings can be seen in the APPENDIX A

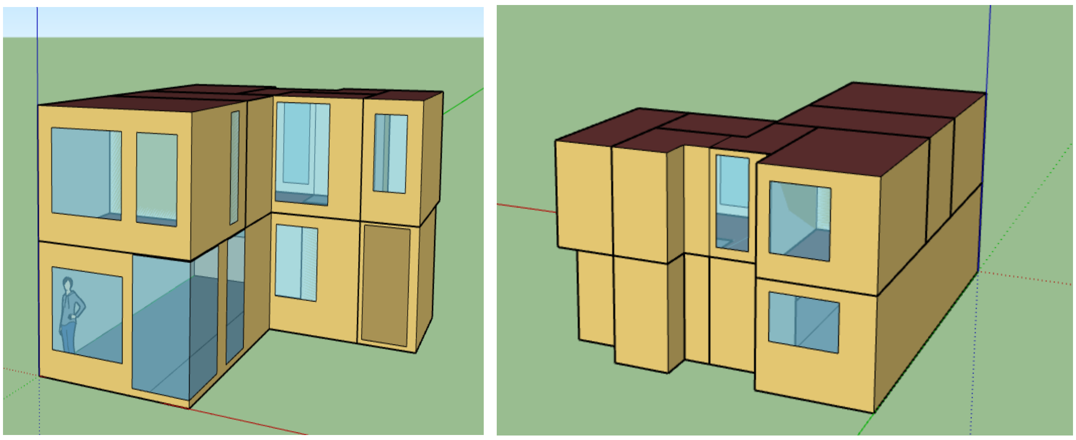


Figure 3.2 Nottingham house model

The next step is that models are imported to EnergyPlus, the parameters for the buildings can be defined by IDF Editor including heating temperature, room occupancy, location (containing weather data), building material properties, internal heat gains and airflow.

Energy demand assessment of the site: Case of Creative Energy Homes (CEHs)

Habitable space can be divided into bedrooms, living room, bathroom, halls, kitchen and toilet according to function. As a result, the space temperature for each room is different. Table 3.4 shows the temperature set point for each room [149, 150].

Table 3.4 Each room temperature set point [149, 150]

Room	Living	Bathroom	Bedroom	Halls	Kitchen	Toilet
Temperature setpoint (℃)	22	20	17	19	17	19

The minimum allowable temperature is 17 °C. The temperature setpoint is during the occupancy of the space. The occupancy schedule is defined according Table 3.5 [151, 152].

Table 3.5 Stay time for residence in bedroom and living room [151, 152]

Time	Bedroom	Living room
Weekdays	22:00-08:00	08:00-09:00 18:00-22:00
Weekends	22:00-08:00	08:00-22:00

In this assessment, the temperature of domestic hot water is set at 49 °C, 43.3 °C for kitchen taps and bathroom respectively while hot water storage should be at a temperature above 60 °C according to Department of Health due to the Legionella and preventable temperature [153, 154].

The energy required for space heating is strongly dependent on building envelope material. The U-value of roof, floor, wall, window, and door for CEHs are summarised in Table 3.6. As can be seen the fabric elements of low energy dwellings have low U-value than the minimum requirement under the building regulation Part L1 [155].

Table 3.6 The U-Value of each element [141, 156-159]

U-Value (W/m^2 k) / House	Roof	Floor	Wall	Window	Door
Mark group	0.17	0.17	0.17	0.71	0.71
Nottingham	0.1	0.1	0.1	1.2	2.19
BASF	0.15	0.1	0.15	1.7	1.5
EON	0.12	0.39	0.69	1.59	3
Tarmac 6	0.1	0.15	0.15	1.42	1.5
Tarmac 4	0.1	0.15	0.19	1.77	1.5
David Wilson	0.25	0.45	0.45	3.3	0.75

In the model, also the internal heat gain (i.e., sensible and latent heat) emitted within the internal space are considered. This include occupants, lighting and electrical equipment. The estimate of the occupancy level in each house was first assumed so that internal heat gains from occupants can be quantified, as shown in Table 3.7.

Table 3.7 The number of occupants in each house [160-163]

House	Mark group	Nottingham	BASF	EON	Tarmac 6	Tarmac 4	David Wilson
Number of Occupants	6	3	4	4	4	4	6

Heat gains due to lights and electric equipment can be estimated based on rated wattage of the appliance. Heat gains shows as indicated in their nameplates. Table 3.8 recapitulates the internal heat gain rates used in the model.

Table 3.8 Internal gains [151, 152]

Description	**Heat gain rate**
People	
75W – maximum sensible gain, 55W – maximum latent gain	
Electrical Equipment	
PCs	55W
Monitors	70W
lighting	12 W/m^2
Kitchen Equipment	
Kettle	500W – sensible gain 315W – latent gain
Fridge	50W – sensible gain 125W – nameplate rating
Freezer	320W – sensible gain 810W – nameplate rating
Dishwasher	1120W – sensible gain 2460W – latent gain 7600W - nameplate rating
Convection oven	293W – sensible gain

Figure 3.3 shows working hours of electric equipment in EnergyPlus schedule, the fraction is the percentage of real working time during working period. The fridge and freezer are assumed to operate all day, of which fraction is 1. Although the dishwasher works from 19:00 to 20:00, the real working hour is 48 minutes, meaning the fraction of 0.8. The working time for convection oven and PC is 45 minutes, 1.5 hours respectively, which mean the fraction is 0.75 and 0.5 respectively. Other equipment working hours are defined in the same way.

Dishwasher	Freezer	Kettle	Covection Oven	Fridge	PC	Light
Fraction	Fraction	Fraction	Fraction	Fraction	Fraction	Fraction
Through:12/31	Through:12/31	Through:12/31	Through:12/31	Through:12/31	Through:12/31	Through:12/31
For: AllDays	For: AllDays	For: AllDays	For: AllDays	For: AllDays	For: AllDays	For: AllDays
Until: 19:00	Until: 24:00	Until: 07:00	Until: 18:00	Until: 24:00	Until: 08:00	Until: 08:00
0	1	0	0	1	0	0
Until: 20:00		Until: 08:00	Until: 19:00		Until: 17:00	Until: 09:00
0.8		0.03	0.75		0	0.3
Until: 24:00		Until: 18:00	Until: 24:00		Until: 19:00	Until: 18:00
0		0	0		0	0
		Until: 19:00			Until: 22:00	Until: 22:00
		0.03			0.5	1
		Until: 24:00			Until: 24:00	Until: 24:00
		0			0	0

Figure 3.3 The working hours for each equipment in EnergyPlus Schedule

The final consideration in EnergyPlus model is air infiltration through doors, windows and building envelope cracks, and ventilation. Mechanical Ventilation with Heat Recovery (MVHR) was installed in Mark group house, which provided fresh filtered air into a building whilst retaining most of the energy that had already been used in heating the building. Earth-to-air heat exchangers (EAHE) was installed in the BASF house, the infiltration was 3.5 ach. Nottingham house, Tarmac 6 and Tarmac 4 house had the Mechanical Ventilation (MV). The ventilation for each house is given in Table 3.9.

Table 3.9 The ventilation for each house [164, 165]

House	Ventilation
Mark group	MVHR system=0.46 ach; Air infiltration=0.5 ach
Nottingham	Air infiltration=0.55 ach; MV=0.11ach
BASF	Infiltration=0.25 ach; EAHE infiltration=3.5 ach (assume 1 hour per day)
EON	Infiltration=0.25 ach;
Tarmac 6	Infiltration=0.08 ach; MV=0.18 ach
Tarmac 4	air Infiltration= 0.14 ach; MV 0.36 ach
David Wilson	Air permeability = $10m^3/h/m^2$

3.5 Results and data analysis

3.5.1 Energy consumption

The dynamic heat demand of the buildings varies according to outdoor air temperature. The annual outdoor air temperature variation in Nottingham is shown in Figure 3.4, the lowest temperature occurred on January 3rd at -2.03 °C, and the highest temperature was 22.85 °C on August 19th. The average temperature in December and January is 4.98 °C, 4.6 °C respectively, while the average temperature in August is 16.27 °C.

The simulation programs avoided using single year, Test Reference Year-type (TRY) weather data. No single year can represent the typical long-term weather patterns. More comprehensive methods that attempt to produce a synthetic year to represent the temperature, solar radiation, and other variables within the period of record are more appropriate and will result in predicted energy consumption and energy costs that are closer to the long-term average.

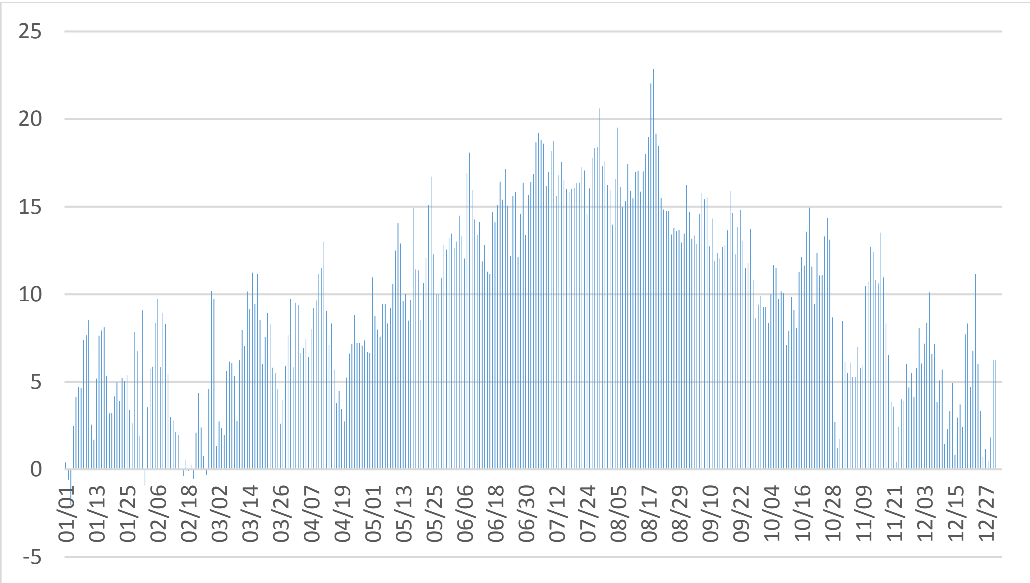


Figure 3.4 Daily outdoor air dry-bulb temperature in Nottingham.

The EnergyPlus energy demand simulation uses the Nottingham house as an example to show the variation about heat load of space heating and domestic hot water during a year. The heat demand for the Nottingham house can be seen in the Figure 3.5, including space heating and domestic hot water. The hot water demand in January was 132.02 kWh, while it was 92.2 kWh in August. However, the building required very little energy for the space heating as it was highly insulated. The amount of energy for space heating demand in January was 402.25 kWh. The hot water heat demand decreased by 30.8% from winter to summer, while the space heating demand decreased by 100%. The space heating demand was mostly from October to April. The space heating demand increased nearly 11 times from October to November, while it decreased by 65.9% from February to March. The maximum space heating demand appeared in January because of low outdoor air temperature.

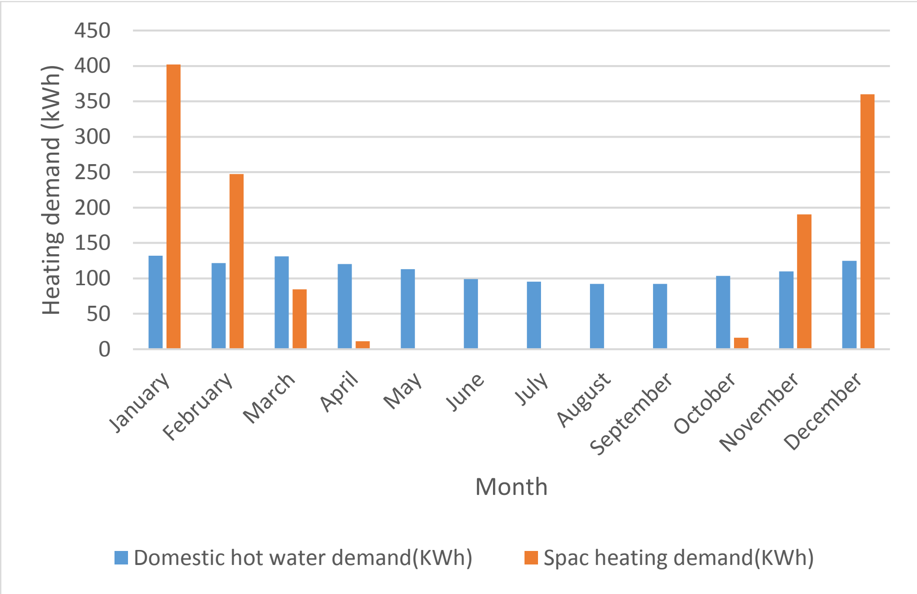


Figure 3.5 The domestic hot water and space heating demand for the Nottingham house

The domestic hot water demand and space heating demand of Mark group, Nottingham, EON, Tarmac 6, Tarmac 4 and David Wilson houses in each month is shown in Table 3.10. There was an increase in space heating requirements from September to January, and a decrease from January to May. This is due to outdoor temperature changes throughout the year and the associated requirement for comfortable indoor temperatures. There is a steady-state for domestic hot water demand throughout a year, the difference of each house is caused by the number of occupants.

Table 3.10 Monthly space and domestic hot water heat demand for creative homes

Date	Mark research (kWh)		Nottingham one (kWh)		BASF (kWh)		EON (kWh)		Tarmac 6 (kWh)		Tarmac 4 (kWh)		David Wilson (kWh)	
	DHW	SH	DHW	SH	DHW	SH	DHW	SH	DHW	SH	DHW	SH	DHW	SH
Jan	264	1223	132	402	177	668	177	972	165	182	165	448	264	2079
Feb	243	978	122	248	162	522	162	794	149	139	149	359	243	1672
Mar	262	483	131	84	175	306	175	488	164	39	164	157	262	1023
Apr	240	200	120	11	160	217	160	307	160	4	160	46	240	605
May	226	22	113	0	151	78	151	78	164	0	164	4	226	204
Jun	198	0	99	0	132	11	132	3	159	0	159	0	198	32
Jul	191	0	96	0	127	0	127	0	165	0	165	0	191	4
Aug	184	0	92	0	123	5	123	2	164	0	164	0	184	34
Sep	184	3	92	0	123	23	123	51	159	0	159	1	184	150
Oct	207	217	103	16	138	147	138	308	165	4	165	51	207	656
Nov	220	783	110	191	147	410	147	666	159	76	159	247	220	1422
Dec	249	1176	125	360	166	608	166	903	165	157	165	411	249	1977

Energy demand assessment of the site: Case of Creative Energy Homes (CEHs)

As Table 3.11 shows, the space heating demand for the Eon, Mark group, David Wilson house are higher than others. Mark group, David Wilson and EON are the top three by the largest total floor area as Table 3.12 shows. It is known that the high floor often constructs a larger room volume, leading to a high space heating requirement. The space heating demand during summer (from June to August) of BASF, EON, David Wilson house is very low and even to 0 for Mark research, Nottingham one, Tarmac 6 and Tarmac 4 house. The domestic hot water demand of each house in winter is higher than in summer due to the outdoor temperature. The hot water depends on the number of occupants, which lead to domestic hot water heat demand.

Table 3.11 Annual energy consumption per house

House	Annual domestic hot water demand (kWh)	Annual Space heating demand (kWh)	Total heating demand (kWh)
Mark group	2669.75	5084.21	7753.96
Nottingham	1334.88	1312.91	2647.79
BASF	1780.75	2996.02	4776.77
Eon	1780.75	4572.43	6353.18
Tarmac 6	1937.50	599.08	2536.58
Tarmac 4	1937.50	1723.38	3660.88
David Wilson	2669.75	9859.90	12529.65
Annual Total	14110.89	26417.92	40258.81

Table 3.12 Total floor area for each house

House	Total floor area (m^2)
Mark group	242.71
Nottingham	79.84
BASF	100.58
Eon	126.12
Tarmac 6	89.2
Tarmac 4	108.7
David Wilson	143.04

The aggregate heat demand of all areas is shown in Figure 3.6 shows. The maximum space heating demand was 5974 kWh in January, while the minimum space heating demand was 4 kWh in July. The maximum hot water heat demand was 1344 kWh in January, the minimum occurred in September, which decreased by 23.8% compared with January.

Changes in domestic hot water demand were small compared with space heating demand. Overall, the maximum heat demand 7318 kWh occurred in winter, while the minimum one is in summer with 1066 kWh. The heat demand decreased by 34.14% from winter (February) to spring (March) and by 28.97% from spring (May) to summer (June), while it increased by 16.47% from summer (August) to Autumn (September). The heat demand nearly doubled from autumn (October) to winter (November), which was the biggest change is this four-season transition.

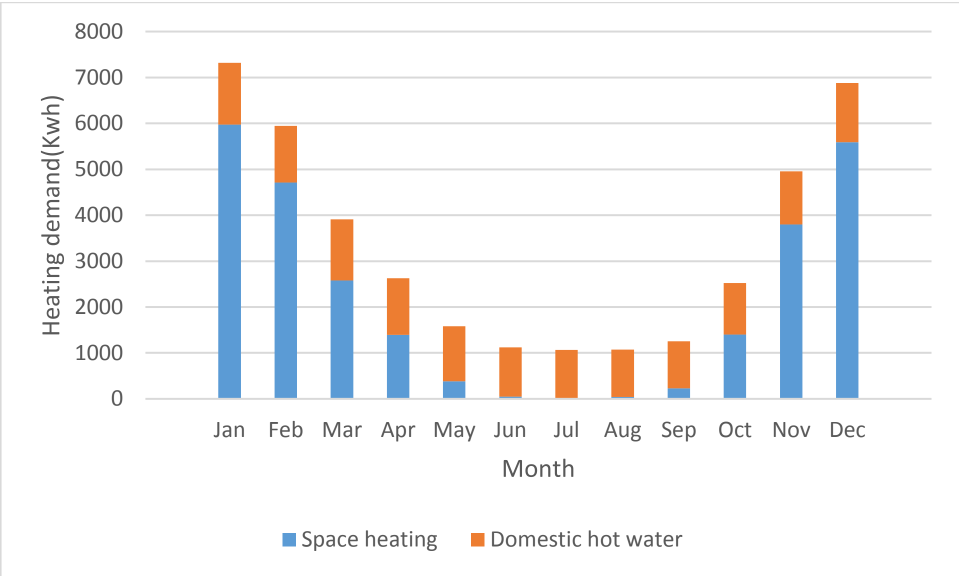


Figure 3.6 Aggregate heat demand for CEHs.

3.5.2 Heat load profile

The hourly heat load for the Nottingham house is given in Figure 3.7. The peak heat load appeared at 8:00 on January 1st, which is 4.14kW. The heat load contains space and domestic heat load. The hourly heat load from April to October was less than the rest month. There was only domestic hot water heat from June to September. Although there was space heating in April, May and October, the space heat demand is very low. The accumulation for hourly heat load during 24 hours can form daily heat demand. The peak load for the Nottingham house can be obtained according to hourly heat load, which is better to choose the size of heat sources.

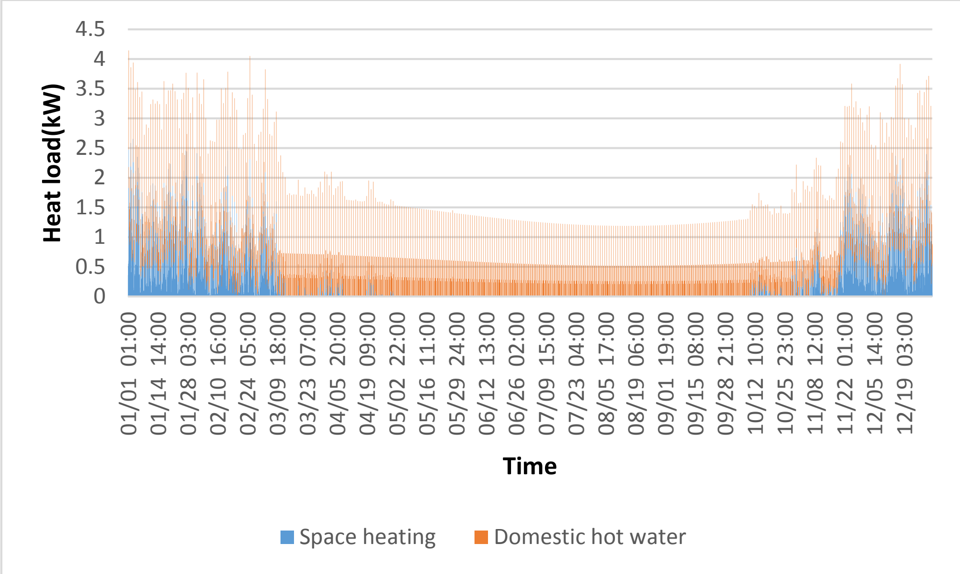


Figure 3.7 Annual space and domestic hot water heat load for the Nottingham house

The peak heat load of other houses is given in Table 3.13, which takes into account of solar and casual heat gains, geometry, thermal mass, occupants, and external temperature variation. The peak load can be used to select heat sources and pipes. The maximum peak load in the seven houses was David Wilson house with 10.9042 kW, while the minimum one was Tarmac 6 house with 3.1818 kW.

Table 3.13 The peak load for each house

House	Peak heat load (kW)
Mark group	9.4757
Nottingham	4.1447
BASF	5.9602
Eon	6.3920
Tarmac 6	3.1818
Tarmac 4	4.1386
David Wilson	10.9042

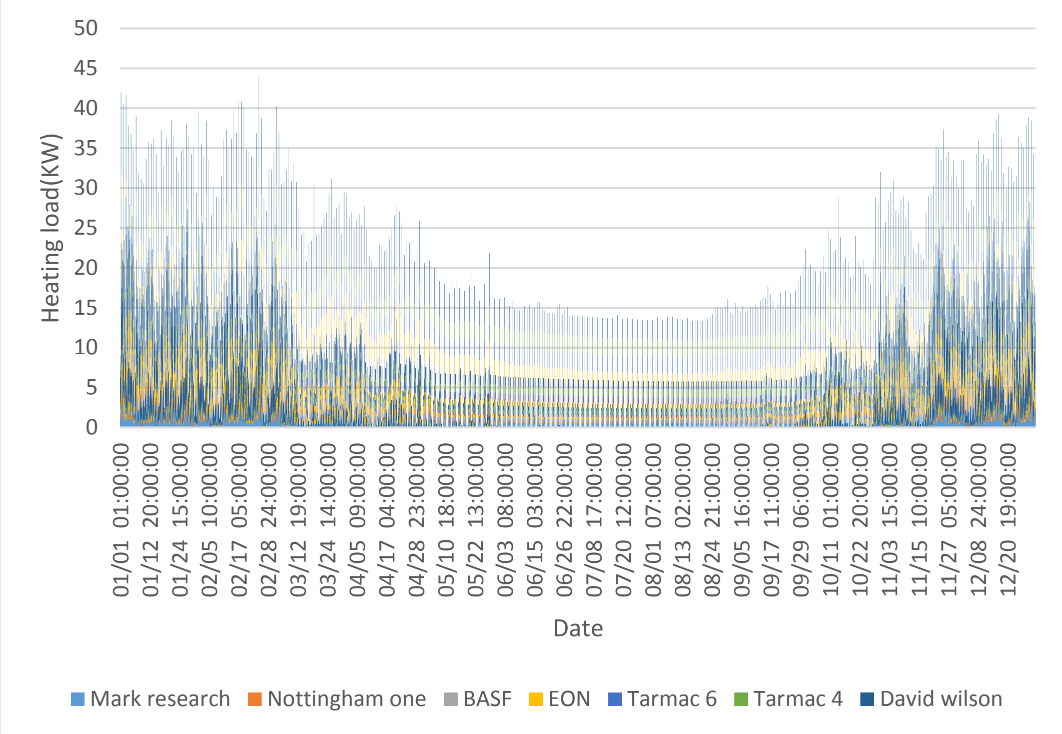


Figure 3.8 Aggregated home heat load for CEHs

In order to control the district heating system, the total heat load for this system throughout a year is given in Figure 3.8, which can be used to select the thermal store and pipes within the system. It shows aggregated home energy consumption containing space and domestic hot water heat load, which is the sum of heat load for the seven buildings. The peak heat load is 44 kW, which is largely less than traditional buildings. Because the buildings are low energy houses and the peak load of each house is not at the same time.

The peak heat load is defined by the hourly heat load, while the hourly heat load is defined by outdoor air temperature and indoor temperature. The variation for indoor temperature is small, the hourly heat load is largely influenced by the outdoor temperature. The week from January 1st to 7th is the average lowest temperature week. The hourly heat load for January 1st to 7th can be seen in Figure 3.9. The daily profiles during this time highlighted two peaks which were related to the typical demand for SH and DHW during mornings and evenings. The profiles for January 1st and January 7th were different from the profiles within January 2nd to 6th, because January 1st and 7th were weekends, while the period from January 2nd to 6th was working days. The difference between weekend heat load pattern and weekday heat load pattern can be obtained by the examples of January 1st (weekend) and January 3rd (weekday). Figure 3.10 and Figure 3.11 present a more detailed insight into weekday and weekend heat load.

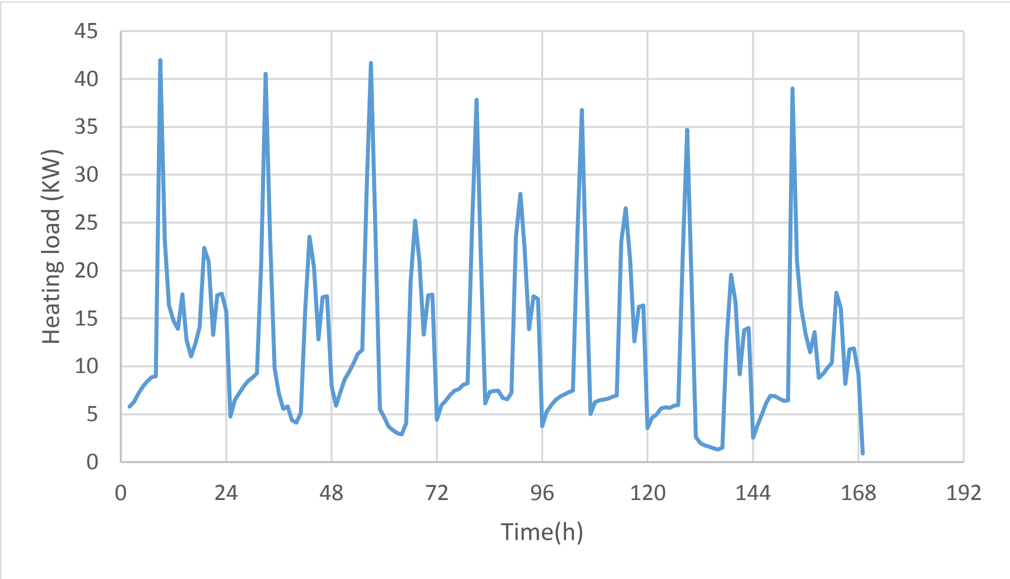


Figure 3.9 Heat load of CEHs for January 1st -7th

In Figure 3.10, the initial peak was around 8:00 leaded due to the sudden requirements of space heating and domestic hot water, for occupants wake up during this time. After the initial peak load, the heat load decreased under the influence of heat gains and occupancy. The next peak load was around 13:00, which was caused by the increase of domestic hot water, because lunch is cooked for hot water requirement during this time. Subsequently, the peak occurred at 18:00 due to the external temperature decrease and a requirement for heat. The heat load increased from 20:00 to 22:00 because of the increase in domestic hot water for bathing and washing.

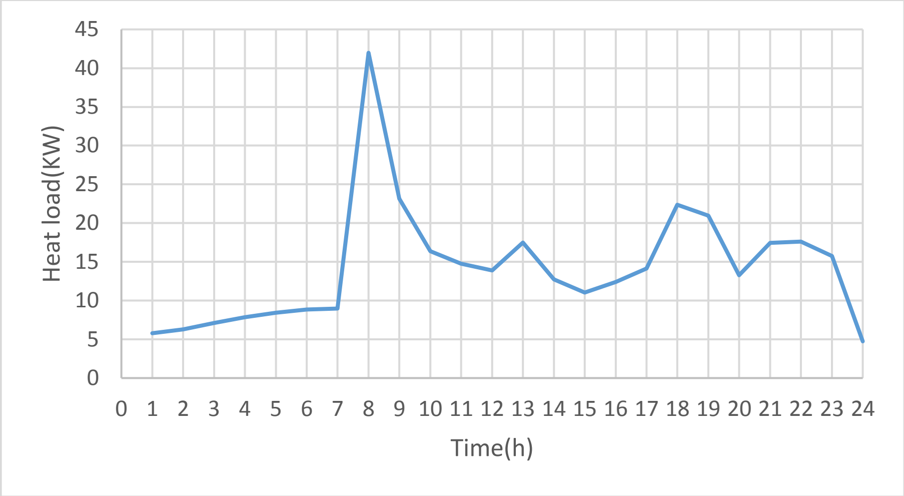


Figure 3.10 Example of standard weekend heat load (January 1st)

In Figure 3.11, the peak heat load was around 08:00 and 18:00, because people leave from home to work from 09:00 to 17:00. As a result, the heat demand was typically lower during this period. The heat load decreased after 18:00, while it reversed from 20:00 to 22:00 due to the increase in domestic hot water for bathing and washing.

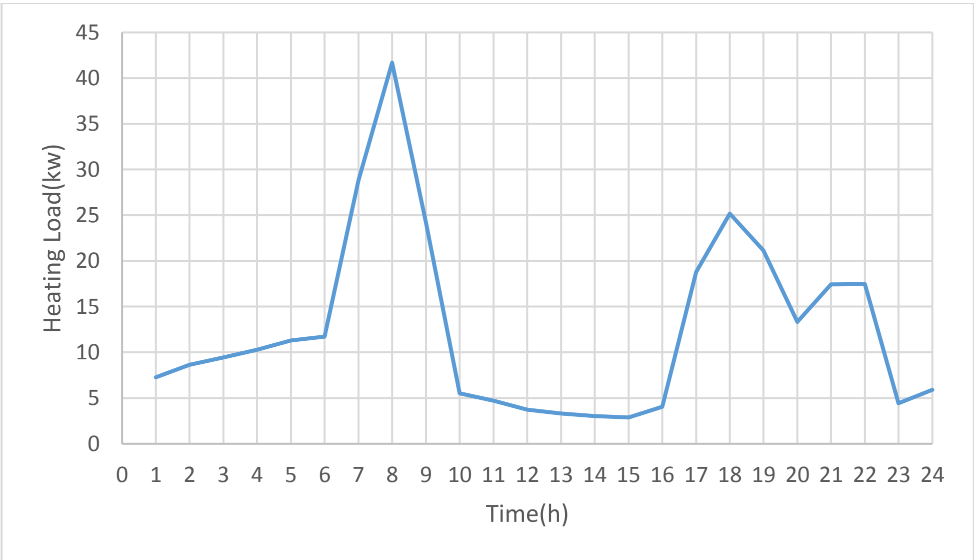


Figure 3.11 Example of standard weekday heat load (January 3rd)

The duration curve of CEHs in Figure 3.12 represents the number of hours in a year at which the load is at or above a particular value, while the absolute site peak heat load in the year is 44 kW. According to Figure 3.8, there are 8760 values in an annual heat load profile. The heat load is ranked in decreasing order, and the abscissa hours is converted to percentage, that is, the 8760 hours is convert to percentage. The 1 hour is converted to 0.01%, while the 8760 hours is converted to 100%. The annual heat load profile can be obtained from Figure 3.8, and the average heat load during a year is 4.16 kW. In Figure 3.12, 4.16 kW corresponds to 33.08% in load duration curve. The above 4.16 kW heat load is considered in the peak demand bracket. Therefore, the peak heat load occurs for around 33.08% of the year corresponding to very low outdoor temperature. The heat load between 4.16 kW and 0 kW accounts for 43.24% of the year and covers the majority of the demand. On the contrary, the site heat load at 0 kW accounts for 23.68% of the year due to no heat demand during the period.

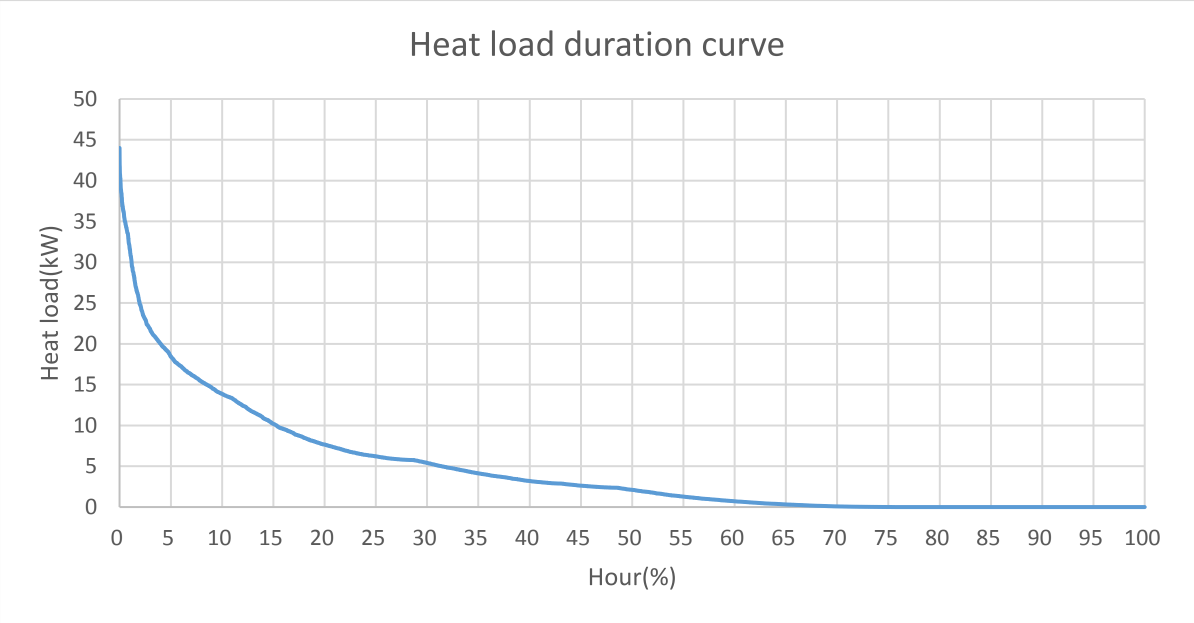


Figure 3.12 CEHs load duration curve

3.6 Summary

The CEHs described in detail from the construction system, efficient heating and cooling system, which provided a unique opportunity to apply LTDH system. Energy Plus software is used to calculate heat of dwellings. The heat energy consumption for each building can be obtained. As a result, the annual energy consumption for the CEHs is 40258.1 kWh, including 14110.89 kWh for domestic hot water and 26417.92 kWh for space heating. The variation in space heating and domestic hot water demand for a year can be obtained. The maximum space heating and domestic hot water demand both occur in January with 5974 kWh and 1344 kWh respectively. Variation in domestic hot water demand with 23.8% is less than in space heating demand during a year. The biggest changes change is this four-season transition from autumn to winter, which increased nearly 1 time for heat demand.

The annual heat load for each building can be obtained, therefore, the annual heat load for CEHs can be acquired. From the heat load profile from January 1st to 7th, the daily profiles during this time highlight two peaks which are related to the typical demand for SH and DHW during mornings and evenings. The weekend heat load profile and weekdays profile are compared, weekend heat load profile has one more peak point than weekdays profile due to the domestic hot water requirement for people during the weekend.

The CEHs duration curve is obtained, the heat load from 15 kW to 44 kW is considered at the peak demand, which account for 33.08% of the year. The heat load between 15 kW and 0 kW accounts for 43.24% of the year and covers the majority of the demand. Conversely, the site heat load at 0 kW accounts for 23.68% of the year.

4 Analysis of the Low Temperature Heat Network heat sources

4.1 Introduction

In this chapter, the multiple heat sources with different outputs are optimised in the LTDH system for a community. The mathematical model is established and then it is input to MATLAB to simulate the hourly variation for LTDH system.

In this study, MATLAB version 9.2 (March 2013 release) was used to simulate the LTDH model. MATLAB is a multi-paradigm numerical computing environment and proprietary programming language developed by MathWorks, which makes the calculation more convenient. Moreover, the results files can be exported to excel, then analysed and discussed.

Firstly, a description of the LTDH system of the site is provided in section 4.2. Then, the mathematical model for LTDH system is established in section 4.3. The mathematical model is conducted in MATLAB to simulate for optimisation of operation of LTDH system. Finally, Section 4.5 provides a summary of this chapter.

4.2 Description of the case study heat network

The CEHs provide an opportunity to realise the low temperature heating system with low energy consumption and carbon in the community. The houses are new built or existing houses which are retrofitted with reasonable constriction and energy saving technology. CEHs have multiple heat sources to supply the heat demand, including solar collector, heat pump, biomass boiler and gas boiler.

The LTDH system for the CEHs (Figure 4.1) offers the flexibility of integrating any type of heat sources regardless of location. The heat source supplies heat to the storage and then the heat stored in the thermal store. Subsequently, the heat from the storage is transported to the end-user through the heat exchanger. This is four-pipe network system. Although the drawback of this is the doubling the cost and heat losses of heat networks. It can fully exploit the ability of heat sources and avoid heat loss of heat sources. The pipe is Rehau's Rauthermex, which offers excellent insulation performance through its PU foam insulation with a lambda value of 0.0216 Wm/k, allowing specifiers and contractors to optimise both installation and operational costs.

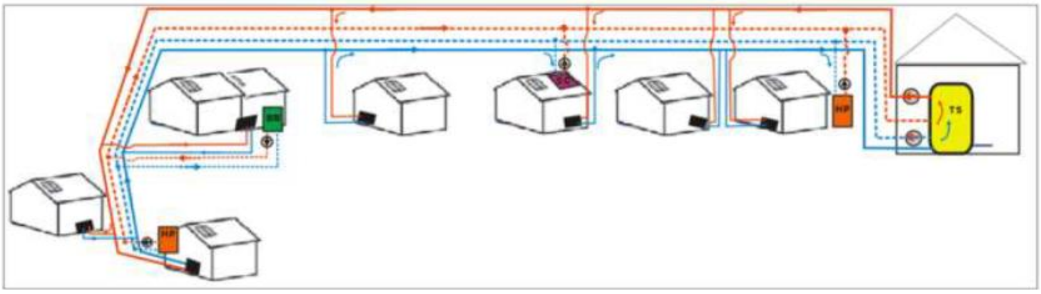


Figure 4.1 The Low Temperature District Heating for CEHs

The optimisation of heat sources of the LTDH is to find the reasonable output of multiple heat sources under three conditions. Firstly, the heat sources can satisfy the heat demand all the time. Secondly, the capacity of thermal store is as small as possible. Lastly, the total output of heat generations is as small as possible

4.3 Methodology

4.3.1 Introduction

The method developed in recent studies to determine the optimum size of a thermal store for district heating is by analysing the heat load from the previous year combining with the reservoir storage allocation analogy in water dam [166]. Optimising the thermal store not only meets the total heat demand for the houses, but also generates an optimal operation schedule for heat generation available for district heating.

It starts to collect annual hourly heat load with a resolution of 8760 values per year, which represents the seasonal heat load characteristic. By assuming the initial heat storage, the minimum storage required, the thermal storage capacity and heat generation needed can be determined using ‘reservoir storage allocation in water dam' analogy.

Furthermore, the reliability of heat generation can be defined by the minimum thermal store capacity required based on a few assumptions. Base on the reliability calculation and graph, the heat generation which cannot achieve heat demand are excluded. Subsequently, the optimal heat generations can be defined by choosing the small generation output at the same storage capacity.

4.3.2 Heat capacity calculation on the annual heat load

In order to determine the optimum thermal store capacity, it is necessary to analyse the heat load variation. The heat load fluctuation per hour can be used to determine the heat required from heat sources and thermal store. In order to calculate the thermal store energy per hour, the analogy of reservoir storage allocation can be applied. However, a few assumptions have to be defined such as minimum thermal store capacity and initial thermal store capacity. The equation can be described as [167]

$$\mathrm{Q}^{\mathrm{Storage}} = \mathrm{Q}^{\mathrm{Initial}} + \mathrm{Q}^{\mathrm{In}} - \mathrm{Q}^{\mathrm{Out}} \qquad (4-1)$$

Where $Q^{Storage}$ is the capacity of the thermal store (kW); $Q^{Initial}$ is the initial thermal store or the minimum storage capacity (kW); Q^{In} is the accumulation of heat supply by multiple heat sources (kW); Q^{Out} is the heat load for the dwellings (kW).

The multiple heat sources can be any combination of fossil fuel and renewables, the Q^{In} can be expressed:

$$\mathrm{Q}^{\mathrm{In}} = \mathrm{Q}^{\mathrm{S}} + \mathrm{Q}^{\mathrm{H}} + \mathrm{Q}^{\mathrm{B}} + \mathrm{Q}^{\mathrm{G}} \qquad (4-2)$$

Where Q^{S} is the heat supplied by solar collector (kW); Q^{H} is the heat supplied by heat pump (kW); Q^{B} is the heat supplied by biomass boiler (kW); Q^{G} is heat supplied by gas boiler (kW). By combining a few renewable energy resources, heat demand can be satisfied in every season throughout the year. Therefore, the equation can be written as

$$Q^{Storage} = Q^{Initial} + Q^{S} + Q^{H} + Q^{B} + Q^{G} - Q^{Out} \qquad (4-3)$$

However, to make the system efficient, the heat supply Q^{In} is not required for every hour. When the heat available inside the thermal store is sufficient to cover the heat load, heat supply is not needed. The storage would be discharge heat based on the demand until the heat of the thermal store almost reaches the minimum thermal store capacity. Therefore, the input heat form heat sources will charge thermal store when the thermal store capacity is not sufficient to meet heat load.

4.3.3 Heat supply

To calculate the heat supply needed for every hour in a year, a few heat supply values have to be assumed. In this case, the heat pump, biomass boiler and gas boiler output assumption will be defined by different sizes. The heat from the solar collector can be calculated by the following equation [168]

$$P_{sh} = \eta \mathrm{GS} \qquad (4-4)$$

$$\eta = \left(\eta_0 K_\theta - \frac{a_1 (T_f - T_a)}{G} - \frac{a_2 (T_f - T_a)^2}{G} \right) \qquad (4-5)$$

Where P_{sh} is heat production of solar collector (W); S is solar collector area (m^2);G is solar radiation on solar collector (W/m^2), T_f is the collector average temperature (°C); T_a is ambient temperature (°C).

The efficiency of the solar collector is defined by three parameters:

η_0: Intercept (maximum) of the collector efficiency.

α_1: The first-order coefficient in collector efficiency equation, (W/m^2°C)

α_2: The second-order coefficient in collector efficiency equation, (W/m^2°C)

For the calculation of the overall solar collector array efficiency, the influence of incidence angle on collector efficiency (K_θ) is needed.

The heat generation from renewable energy resources and fossil fuels which generates heat to supply the demand. The excessive heat will be stored in the thermal storage which can be consumed for future demand. The calculation for heat supply is correlated with the thermal storage allocation. When the storage level cannot reach the minimum storage required, heat generation will start to work to supply heat. When the storage level is above minimum storage required, then the heat supply will be zero.

The working period of each heat source varies greatly in a day and a month. The peak heat load usually appears in the winter, all the heat sources need to work to supply the peak load. Even though the heat load during the summer is low, the heat supply from solar collector is not always sufficient to meet demand. The working order of each heat source can be prioritised according to fuel cost and carbon emission intensity. The priority for each heat source can be set up as Table 4.1 shows.

Table 4.1 The priority of each heat source

Month	Time	Solar collector	heat pump	biomass boiler	Gas boiler
Jan-May	00:00-6:59	1	2	3	4
	07:00 -23:59	1	3	2	4
Jun-Sep	00: 00-6:59	1	2	3	4
	07:00 -23:59	1	2	3	4
Oct-Dec	00: 00-6:59	1	2	3	4
	7:00 -23:59	1	3	2	4

As a first priority (number 1) was attributed to the solar collector as it is renewable source of energy with no fuel cost and no direct emission. The gas boiler on the other hand was least desirable heating system of heat energy mix and was assigned a priority number of 4. The heat pump (priority order 2 or 3) and biomass boiler (priority order 2 or 3) working order varies according to time and month of the year.

Taking into account the working order of the appliances, the hourly heat balance for the thermal store of Equation (4-3) can be rewritten as follows

$$Q_{i,j+1}^{Storage} = Q_{i,j}^{Storage} + Q_{i,j+1}^{S} + \alpha Q_{i,j+1}^{H} + \beta Q_{i,j+1}^{B} + \gamma Q_{i,j+1}^{G} - Q_{i,j+1}^{out} \qquad (4-6)$$

Where indices i, j represent day and hour respectively. The parameter of α, β, γ take the value of '1' or '0' depending if the system is on or off respectively. This is subject to the following operating conditions:

$$Q_{i,j+1}^{Storage} \gg Q^{Intial}$$

$$Q^{Initial} = Q_{i,j}^{Storage} \text{ For i=1, j=0}$$

The thermal store capacity always cannot less than the minimum storage capacity. The initial storage capacity is assumed to equal to the minimum storage capacity.

With day 1 is taken as January 1st, the range of day 'i' from 1 to 151 are from January 1st to May 31st and 'i' from 274 to 365 represent from October 1st to December 31st. For the hour of day 'j' ranging from 0:00 to 07:00, all the heat sources are set to idle ($\alpha = 0, \beta = 0, \gamma = 0$) and the heat demand is met by the thermal store. However, the residual thermal store's heat capacity is less than initial thermal store capacity, the heat pump ($\alpha = 1$) starts to work for one hour to supply the rest heat, while the biomass boiler and gas boiler remain idle ($\beta = 0, \gamma = 0$). After that, if the thermal store capacity still less than minimum thermal store capacity, then the biomass boiler is turned on alongside the heat pump ($\alpha = 1, \beta = 1$). After that, if the residual thermal store capacity is still less than minimum storage capacity, the gas boiler is started to provide peak load in addition to the heat pump and biomass boiler ($\alpha = 1, \beta = 1, \gamma = 1$).

For day 'i' from 1 to 151 or i from 274 to 356 and j from 0 to 7

$$\alpha = 0, \beta = 0, \gamma = 0$$

If

$$Q_{i,j+1}^{Storage} \leq Q^{Intial}$$

$$\alpha = 1, \beta = 0, \gamma = 0$$

Then if

$$Q_{i,j+1}^{Storage} \leq Q^{Intial}$$

$$\alpha = 1, \beta = 1, \gamma = 0$$

If $Q_{i,j+1}^{Storage} \leq Q^{Intial}$

$$\alpha = 1, \beta = 1, \gamma = 1$$

For the rest of the day hours (i.e., ‘j’ from 08:00 to 23:00), all the heat sources are turned off ($\alpha = 0, \beta = 0, \gamma = 0$), if the thermal store can supply the heat demand. If the residual heat capacity of thermal storage is less than the minimum storage capacity (initial thermal storage capacity), the biomass boiler is switched on ($\beta = 1$) for one hour to supply the required heat, while the heat pump and gas boiler remain switched off ($\alpha = 0, \gamma = 0$). After that, if the residual thermal storage capacity is still less than minimum storage capacity, then the heat pump is turned on alongside the biomass boiler ($\alpha = 1, \beta = 1$), while the gas boiler remains the idle state ($\gamma = 0$). After that, the thermal storage capacity still less than minimum storage capacity, the gas boiler is started to provide peak load in addition to the heat pump and biomass boiler ($\alpha = 1, \beta = 1, \gamma = 1$).

For day hour of ‘j’ from 8: 00 to 23:00

$$\alpha = 0, \beta = 0, \gamma = 0$$

If

$$Q_{i,j+1}^{Storage} \leq Q^{Intial}$$

$\alpha = 0, \beta = 1, \gamma = 0$

Then if

$$Q_{i,j+1}^{Storage} \leq Q^{Intial}$$

$$\alpha = 1, \beta = 1, \gamma = 0$$

If $Q_{i,j+1}^{Storage} \leq Q^{Intial}$

$$\alpha = 1, \beta = 1, \gamma = 1$$

From June to September (i.e., 'i' from 152 to 273), all the heat sources are turned off ($\alpha = 0, \beta = 0, \gamma = 0$), if the thermal store can supply the heat demand. However, the residual thermal store's heat capacity is less than initial thermal store capacity, the heat pump ($\alpha = 1$) starts to work for one hour to supply the rest heat, while the biomass boiler and gas boiler remain idle ($\beta = 0, \gamma = 0$). After that, if the thermal store capacity still less than minimum thermal store capacity, then the biomass boiler is turned on alongside the heat pump ($\alpha = 1, \beta = 1$). After that, if the residual thermal store capacity is still less than minimum storage capacity, the gas boiler is started to provide peak load in addition to the heat pump and biomass boiler ($\alpha = 1, \beta = 1, \gamma = 1$).

For 'i' from 152 to 273, ∀ j

$$\alpha = 0, \beta = 0, \gamma = 0$$

If $Q_{i,j+1}^{Storage} \leq Q^{Intial}$

$$\alpha = 1, \beta = 0, \gamma = 0 \ .$$

Then if

$$Q_{i,j+1}^{Storage} \leq Q^{Intial}$$

$$\alpha = 1, \beta = 1, \gamma = 0$$

If $Q_{i,j+1}^{Storage} \leq Q^{Intial}$

$$\alpha = 1, \beta = 1, \gamma = 1$$

The storage capacity for water tank as follows

$$Q_{storage} = \max\left(Q_{i,j+1}^{Storage}\right), \forall \text{ i, j}$$

The capacity of thermal store is evaluated based on the peak capacity requirement by the thermal store to satisfy the site's heat demand profiles over one year (8760 hours). The mathematical formulation of the system design is implemented in MATLAB, which detail code is shown in Appendix B.

4.3.4 Reliability analysis

Once the storage capacity and the heat supply have been defined for each hour, the reliability analysis has to be calculated. The reliability is expressed with one percentage value for every heat storage capacity and heat supply assumption. Before the reliability is calculated, the 'demand met' has to be determined first by using one and zero value. The thermal store has a residual minimum storage capacity, which cannot be used to supply the heat demand. For every hour, if the thermal store capacity is bigger than the sum of the minimum storage capacity and heat demand, the 'demand met' value is one. Otherwise, if the thermal store capacity does not satisfy the sum of heat demand and minimum storage capacity, the 'demand met' value is zero. The 'demand met' value is expressed for 8760 value per year. Therefore, the reliability can be calculated as follows:

$$Reliability = \frac{\sum_{n=1}^{8760} demand\ met}{8760} \qquad (4-7)$$

From the Equation (4-7), the reliability graph and table can generate from each building or aggregated buildings heat load profiles in order to determine the sizes of multiple heat sources. If the thermal store can satisfy the heat demand throughout the year, the reliability value would be 100%.

4.4 Results and analysis

4.4.1 Data and assumption

In order to develop an optimisation model of LTDH system, a significant amount of data needed to be gathered and some assumption has to be made. It contains the explanation of the assumption which is taken to determine the optimisation model as well as the possible cases which are developed to analyse the most efficient size for heat sources. In this study, the data provided are only heat load data and current heat supply from a few ranges of energy resources such as gas boiler, heat pump, biomass boiler, and solar thermal collector. Therefore, data assumptions such as a value of heat generation to make the system more efficient, thermal store working temperature, and initial storage volume needed to be assumed.

The output of solar collector can be defined by the Equation (4-4). Solar collector was installed in the buildings, the model is V 30 [169]. The specific performance parameter can be seen in Table 4.2.

Table 4.2 Performance Specification [169]

Model	Unit	V30
Aperture area	m^2	3.1
Zero loss efficiency h_0	____	0.76
Heat loss efficient α_1	W/m^2K	3.6
Second order α_2	W/m^2K^2	0.0068
I.A.M k_θ	____	0.93
Effective heat capacity C_{eff}	kJ/m^2K	5.3
Length	m	2.892
Width	m	1.167

The parameter for the solar collector can be obtained from Table 4.2 and the hourly ambient temperature during a year also needed to calculate the solar energy, which can be obtained from Figure 4.2.

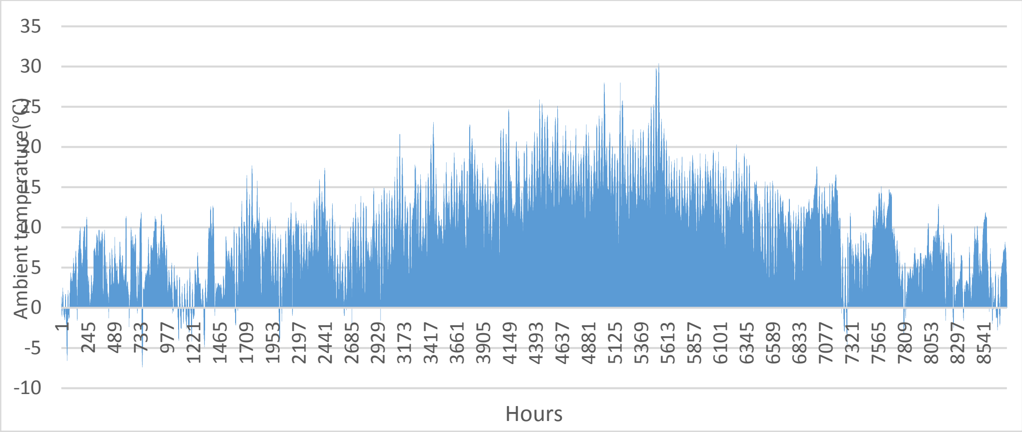


Figure 4.2 Hourly ambient temperature for a year

The inlet/outlet temperature of solar collector is 60/45 ℃ and the global horizontal radiation can be seen in Figure 4.3. As a result, the efficiency of solar collector can be calculated according to Equation (4-5). The output of solar collector can be defined by the area of solar collector. The area of solar collector V30 is 3.1 m^2. The annual solar energy of V30 can be seen in Figure 4.4, there is little solar energy in December and January, while the solar energy was at a low level in February and November. The solar energy at high level focused on July and August above 1.4 kW. The maximum output of solar energy is 1.56 kW.

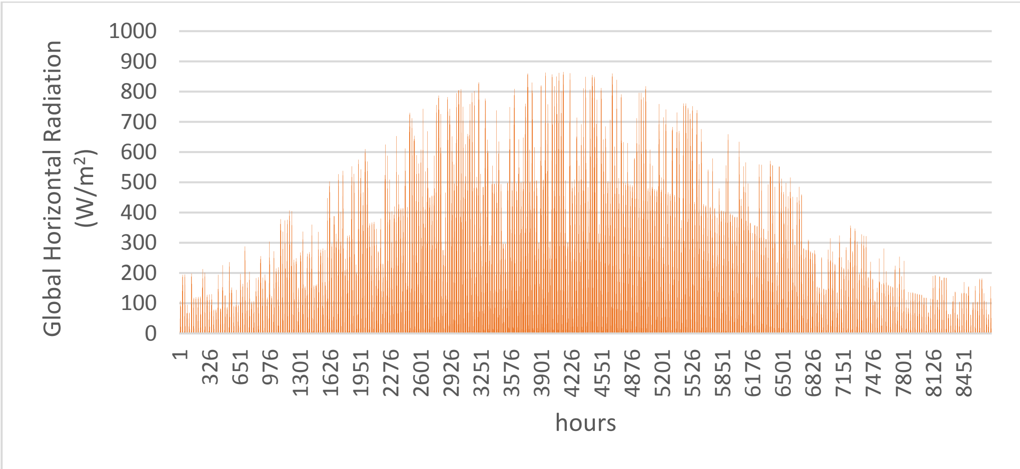


Figure 4.3 Global horizontal radiation for a year

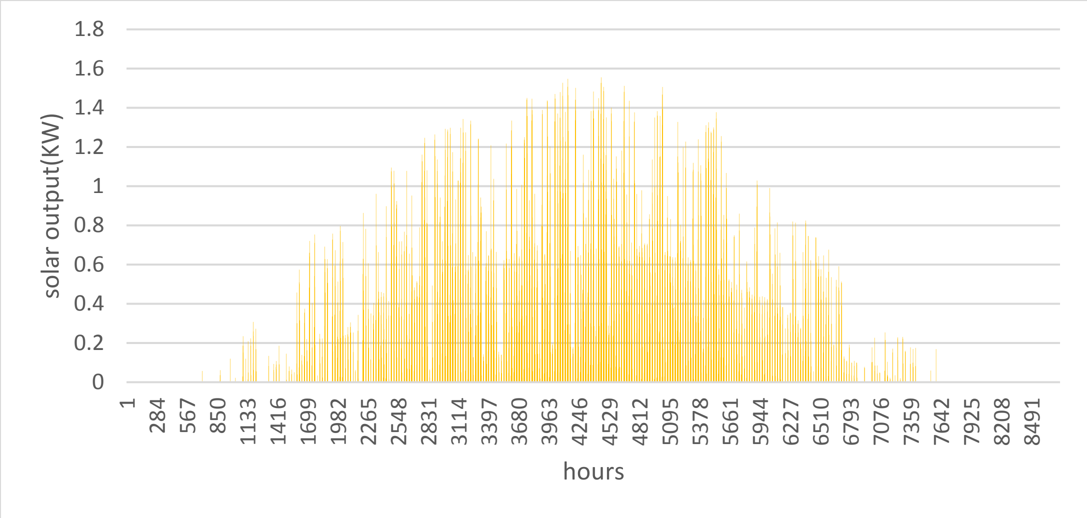


Figure 4.4 Hourly solar output during a year

Three different surface area of solar collectors were considered in the optimisation of heat generation sources: 3.1 m^2, 6.2 m^2, 9.3 m^2 with heat output capacity of 1.56 kW, 3.11 kW and 4.67 kW respectively. Similarly, for the optimisation of the heat pump the following heating capacity were considered: 5 kW, 10 kW, 15 kW, 20 kW, 25 kW and 30 kW while for the biomass boiler be of 5 kW, 10 kW, 15k W, 20 kW, 25 kW, 30kW and 35 kW to ensure that. The peak load is always satisfied with the output of the gas boiler can be assumed 5 kW, 10 kW, 15 kW, 20 kW and 25 kW. The project was to optimize and design the multiple heat sources, the output of each heat source should not be equal 0. In addition, if the gas boiler equals 0, the output of heat pump and biomass boiler would to increase to meet the peak heat load which was not always occurred. The initial invest increased and working efficiency decreased for heat pump and biomass boiler. The power increase in steps of 5kW for heat pump, biomass boiler and gas boiler were more in line with the choice of different outputs in manufactural data. The simulation scenarios for all the different size of heat sources is summarised in Table 4.3.

Table 4.3 Different size of heat sources

MAX output(kW)			
solar collector	Heat pump	Gas boiler	Biomass boiler
1.56(3.1m^2)	5	5	5
3.11(6.2m^2)	10	10	10
4.67(9.3m^2)	15	15	15
	20	20	20
	25	25	25
	30		30
			35

As Table 4.3 shows, the output of solar collector, heat pump, gas boiler and biomass boiler are variable, as a result there are multiple permutations and combinations. In the simulation, a number of scenarios referred to here as the A, B, C, D, E, F were considered. In each of the scenarios, the heat pump heating capacity is fixed to 5 kW, 10kW, 15 kW, 20 kW, 25 kW or 30 kW, while the outputs of solar collector, gas boiler and biomass boiler are allowed to vary.

4.4.2 Scenario A: 5 kW heat pump

The heat pump is fixed at 5 kW, other heat generation outputs are allowed to vary. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW respectively. For example, for 1.56 kW solar collector and a heat pump of 5 kW, the optimum hot water storage capacity is calculated according to Equation (4-3). The numerical results of the simulation are tabulated in Table 4.4.

Table 4.4 The storage capacity for 1.56kw solar collector in the different output of biomass boiler and gas boiler

Gas boiler(kW) / Biomass boiler(kW)	5	10	15	20	25
5	19.5449	21.6780	26.6780	31.6780	36.6780
10	19.8516	19.5449	23.1135	28.1135	33.1135
15	23.9248	23.9248	23.9248	24.9905	29.9905
20	28.8054	28.8054	28.8054	28.8054	29.9905
25	35.2237	35.2237	35.2237	35.2237	35.2237
30	40.3949	40.3949	40.3949	40.3949	40.3949
35	45.3949	45.3949	45.3949	45.3949	45.3949

Table 4.4 shows the thermal storage heat capacity for different output of biomass and gas boiler. It is shown that not all biomass boiler and gas boiler capacity can fulfil the peak heat demand. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was considered as described by Equation (4-7). Table 4.5 shows the results of the analysis where a value of '1' means the combined heat generation capacity and thermal store can satisfy the hourly heat demand reliably throughout the year (i.e. 8760h).

Table 4.5 The reliability for all type of biomass boiler and gas boiler

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	25
5	0.8729	0.9545	0.9832	0.9953	0.9981
10	0.9535	0.9814	0.9939	0.9981	0.9999
15	0.9831	0.9939	0.9984	0.9998	1
20	0.9939	0.9984	0.9999	1	1
25	0.9984	0.9999	1	1	1
30	1	1	1	1	1
35	1	1	1	1	1

For example, when combing a heat generating capacity of a 5 kW gas boiler, 5 kW biomass boiler, 5 kW heat pump and 1.56 kW solar collector, the heat demand can only be satisfied for 87.29% of the time (i.e., there are 1113 hours of the year where the heat demand is not met). Furthermore, Figure 4.5 shows that the reliability increased as the heat generation output of the biomass boiler or/and gas boiler are increased. It is also shown that the reliability of the system reaches '1' for a maximum biomass boiler capacity of the 25 kW and a minimum gas boiler output of 5 kW or vice versa.

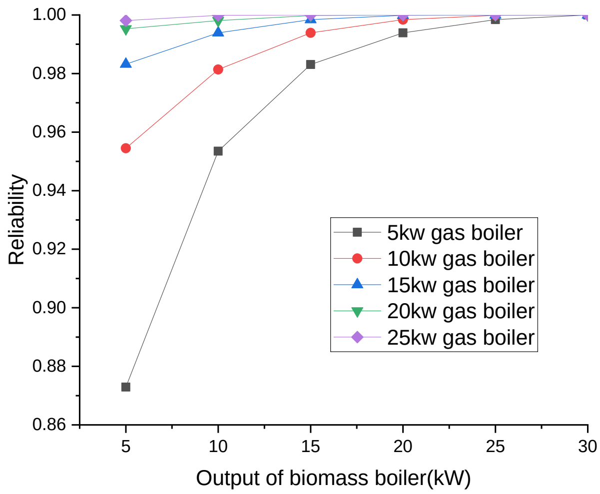


Figure 4.5 The reliability for all type of biomass boiler and heat boiler with 1.56 Kw solar collector

The results of the analysis of the thermal store capacities that can fulfil the heat demand of the site are further presented in the Table 4.6. The heat capacities of the thermal store that provides a reliability of energy supply less '1' have been discarded in this instance.

Table 4.6 The storage capacity for different output of biomass and gas boiler with 1.56 kW solar collector.

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	25
5	----------	----------	----------	----------	----------
10	----------	----------	----------	----------	----------
15	----------	----------	----------	----------	29.9905
20	----------	----------	----------	28.8054	29.9905
25	----------	----------	35.2237	35.2237	35.2237
30	40.3949	40.3949	40.3949	40.3949	40.3949
35	45.3949	45.3949	45.3949	45.3949	45.3949

Similar analytical procedure was repeated for 3.11 kW (6.2 m^2) and 4.67 kW (9.3 m^2) heat output capacity. The results of the analysis are recapitulated in Table 4.7 and Table 4.8 respectively. It can be seen in Table 4.7 and Table 4.8 that more boilers outputs can fulfil the heat demand of the site.

Table 4.7 The storage capacity for different output of biomass and gas boiler with 3.11 kW solar collector.

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	25
5	----------	----------	----------	----------	----------
10	----------	----------	----------	----------	39.8036
15	----------	----------	----------	35.2600	35.2600
20	----------	----------	----------	41.2319	41.2319
25	----------	----------	44.8914	44.8914	44.8914
30	49.2218	49.2218	49.2218	49.2218	49.2218
35	53.7188	53.7188	53.7188	53.7188	53.7188

Table 4.8 shows that increasing the solar thermal collector capacity to 4.67 kW has increased the storage capacity of the system without improvement in the flexibility of boiler rating required.

Table 4.8 The storage capacity for different output of biomass and gas boiler with 4.67 kW solar collector

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	25
5	----------	----------	----------	----------	----------
10	----------	----------	----------	----------	44.2083
15	----------	----------	----------	46.7173	46.7173
20	----------	----------	----------	51.7173	51.7173
25	----------	51.7150	51.7150	51.7150	51.7150
30	59.3592	59.3592	59.3592	59.3592	59.3592
35	66.7173	66.7173	66.7173	66.7173	66.7173

Table 4.6-4.8 shows thermal store capacity for reasonable biomass boilers and gas boilers with 3.1 m^2, 6.2 m^2, and 9.3 m^2 solar collector. The thermal store capacity with different heat sources at a fixed 5 kW heat pump can be seen in Figure 4.6.

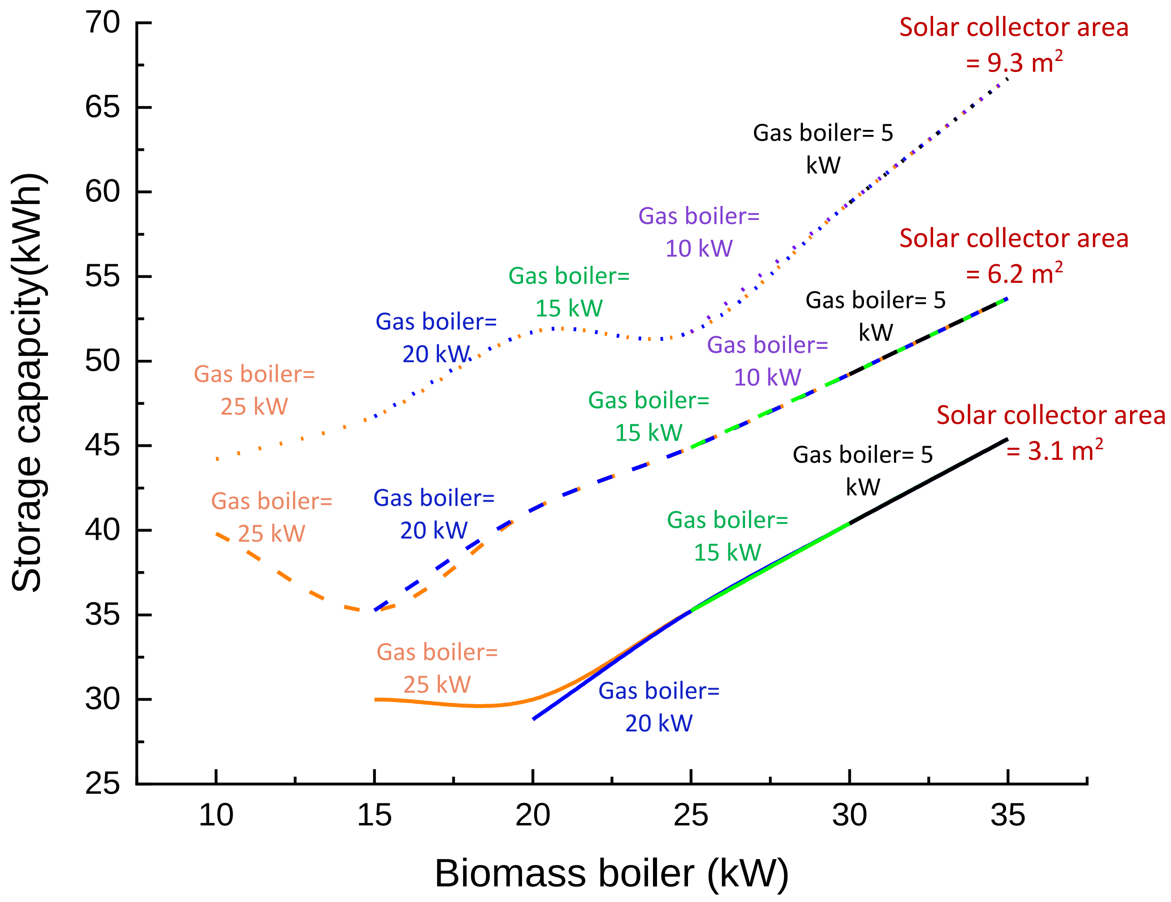


Figure 4.6 Scenario A: Storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 5 kW heat pump

Figure 4.6 shows the thermal store capacity with different gas boiler, biomass boiler and solar collector at a fixed 5 kW heat pump. When the lowest capacity is same at different biomass boiler and gas boiler heat capacities, the priority choice is that gas boiler heat capacity is as small as possible due to fuel cost and carbon emission intensity. Then, the sum of the heat capacity of biomass boiler and gas boiler is the smallest.

The lowest storage capacity with a 3.1 m^2 solar collector is 28.81 kWh, while both gas boiler and biomass boiler are rated at 20 kW. For a solar collector of 6.2 m^2, the lowest storage capacity is 35.26 kWh with a lower biomass boiler and gas boiler rating of 15 kW and 20 kW respectively. Similarly, the lowest storage capacity of a 9.3 m^2 solar collector is 44.21 kWh, while gas boiler and biomass boiler are rated at 25 kW and 10 kW respectively.

4.4.3 Scenario B: 10 kW heat pump

The storage capacity results with different heat generations at fixed 5 kW heat pump have been obtained in Scenario A. Similar analytical procedure was repeated for Scenario B. The heat pump is fixed at 10 kW, other heat generation outputs are allowed to vary. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW respectively. The numerical simulation results for 1.56 kW, 3.11 kW and 4.67 kW solar collectors are tabulated in Table 4.9.

Table 4.9 The thermal store capacity for different output of biomass, gas boiler and solar collector at fixed 10 kW heat pump

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	24.5449	24.5449	24.5449	27.1004	32.1004	1.56
10	23.8026	23.8026	24.9796	29.9561	29.9561	
15	26.678	26.678	26.678	26.678	29.9669	
20	29.5449	29.5449	29.5449	29.5449	29.5449	
25	353949	35.3949	35.3949	35.3944	35.3949	
30	41.678	39.5449	41.678	39.5449	41.678	
5	34.2218	34.2218	34.2218	34.2218-	36.8983	3.11
10	33.1246	33.1249	33.1249	33.1249	33.2472	
15	34.8914	34.8914	34.8914	34.8914	34.8914	
20	40.26	40.26	40.26	40.26	40.26	
25	43.1743	43.1743	43.1743	43.1743	43.1743	
30	50.2600	50.2600	50.2600	50.2600	50.2600	
5	42.5403	42.5403	42.5403	42.5403	43.7728	4.67
10	42.5403	42.9189	42.5403	42.9189	44.2083	
15	42.5403	42.5403	42.5403	42.5403	42.5403	
20	49.3592	47.5403	49.3592	47.5403	49.3592	
25	55.9537	55.9537	55.9537	55.9537	55.9537	
30	57.5403	57.5403	57.5403	57.5403	57.5403	

Table 4.9 shows the thermal storage heat capacity for different output of biomass, gas boiler and solar collector at a fixed 10 kW heat pump. It is shown that not all biomass boiler and gas boiler capacity can fulfil the peak heat demand. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was repeated for 10 kW heat pump as Table 4.10 shows

Table 4.10 The reliability for all type of biomass boiler, heat boiler and solar collector at fixed 15 kW heat pump

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	0.9547	0.9824	0.9950	0.9985	1	1.56
10	0.9826	0.9941	0.9988	0.9999	1	
15	0.9957	0.9986	0.9999	1	1	
20	0.9990	0.9999	1	1	1	
25	0.9999	1	1	1	1	
30	1	1	1	1	1	
5	0.9559	0.9833	0.9946	0.9990	1	3.11
10	0.9831	0.9942	0.9988	1	1	
15	0.9955	0.9991	1	1	1	
20	0.9989	1	1	1	1	
25	1	1	1	1	1	
30	1	1	1	1	1	
5	0.9539	0.9829	0.9950	0.9991	1	4.67
10	0.9833	0.9942	0.9991	1	1	
15	0.9949	0.9990	1	1	1	
20	0.9991	1	1	1	1	
25	1	1	1	1	1	
30	1	1	1	1	1	

The results of the analysis of the thermal store capacities that can fulfil the heat demand of the site are further presented in the Table 4.11. The heat capacities of the thermal store that provides a reliability of energy supply less ‘1’ have been discarded in this instance.

Table 4.11 The storage capacity for Scenario B with different heat generations

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	----------	----------	----------	----------	----------	1.56
10	----------	----------	----------	----------	29.9561	
15	----------	----------	----------	26.678	29.9669	
20	----------	----------	29.5449	29.5449	29.5449	
25	----------	35.3949	35.3949	35.3944	35.3949	
30	41.678	39.5449	41.678	39.5449	41.678	
5	----------	----------	----------	----------	36.8983	3.11
10	----------	----------	----------	33.1249	33.2472	
15	----------	----------	34.8914	34.8914	34.8914	
20	----------	40.2600	40.26	40.2600	40.2600	
25	43.1743	43.1743	43.1743	43.1743	43.1743	
30	50.2600	50.2600	50.2600	50.2600	50.2600	
5	----------	----------	----------	----------	43.7728	4.67
10	----------	----------	----------	42.9189	44.2083	
15	----------	----------	42.5403	42.5403	42.5403	
20	----------	47.5403	49.3592	47.5403	49.3592	
25	55.9537	55.9537	55.9537	55.9537	55.9537	
30	57.5403	57.5403	57.5403	57.5403	57.5403	

Table 4.11 shows thermal store for reasonable biomass boilers and gas boilers with 3.1 m^2, 6.2 m^2, and 9.3 m^2 solar collector. It shows that increasing the solar thermal collector capacity has increased the storage capacity of the system without improvement in the flexibility of boiler rating required. The storage capacity with different heat sources at a fixed 10 kW heat pump can be seen in Figure 4.7.

The lowest storage capacity with 3.1 m^2 solar collector is 26.68 kWh, while the gas boiler and biomass boiler rated at 20 kW and 15 kW respectively. For a solar collector of 6.2 m^2, the lowest storage capacity is 33.12 kWh with a lower biomass boiler and gas boiler rating of 10 kW and 20 kW respectively. Similarly, the lowest storage capacity of 9.3 m^2 solar collector is 42.54 kWh, while gas boiler and biomass boiler heating capacities are reduced further to 15 kW each.

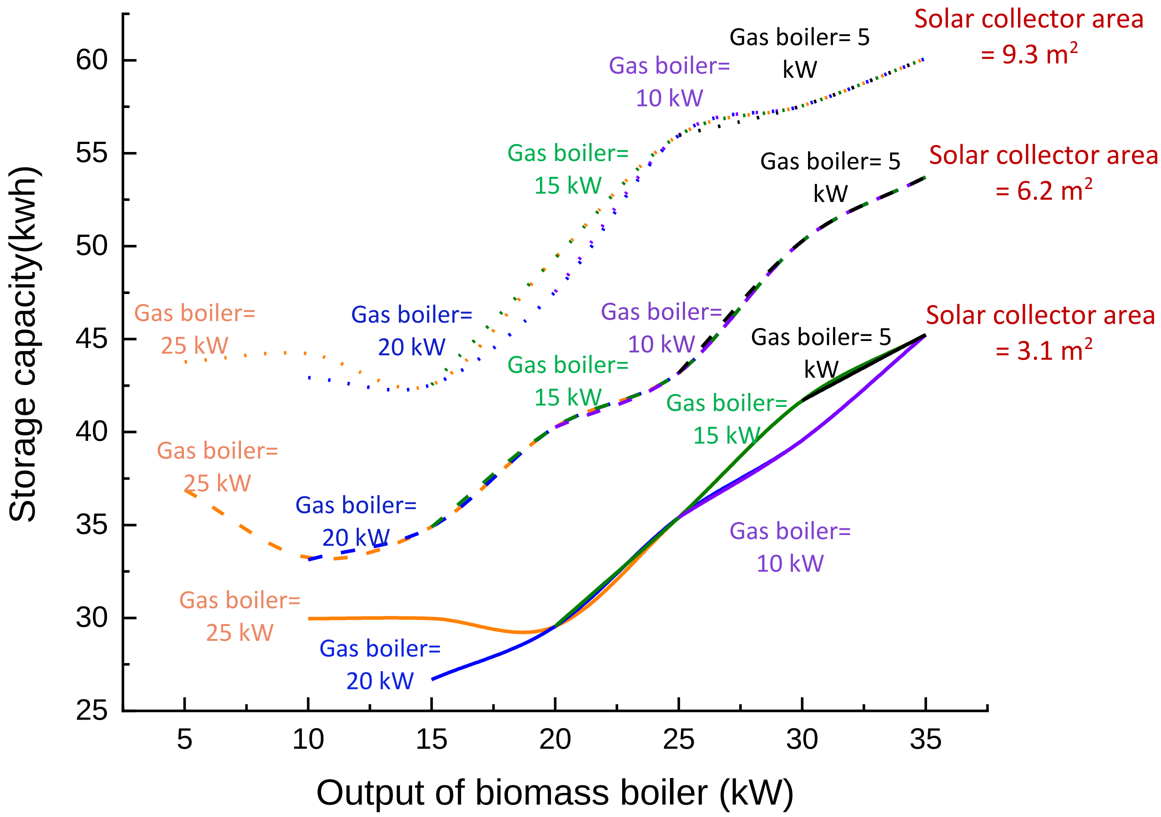


Figure 4.7 Scenario B: Storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 10 kW heat pump

4.4.4 Scenario C: 15 kW heat pump

Similar analytical procedure was repeated for Scenario C The heat pump is fixed at 15 kW, other heat generation outputs are allowed to vary. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW respectively. The numerical simulation results for 1.56 kW, 3.11 kW and 4.67 kW solar collectors are tabulated in Table 4.12.

Table 4.12 The thermal store capacity for different output of biomass, gas boiler and solar collector at fixed 15 kW heat pump

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	29.5449	29.5449	29.5449	29.5449	29.9905	1.56
10	29.5449	29.5449	29.5449	29.5449	29.6967	
15	28.8026	29.5449	29.4454	28.8026	29.8974	
20	29.8516	29.8516	29.8516	29.8516	29.8974	
25	34.5406	34.5406	34.5406	34.5406	34.5406	
30	38.9248	38.9248	38.9248	38.9248	38.9248	
5	38.1743	38.1743	38.1743	38.1743	38.1743	3.11
10	38.1743	38.1743	38.1743	38.1743	38.1743	
15	39.2218	39.2218	39.2218	39.2218	39.2218	
20	38.1743	38.1743	38.1743	38.1743	38.1743	
25	42.0642	42.0642	42.0642	42.0642	42.0642	
30	47.0642	47.0642	47.0642	47.0642	47.0642	
5	46.7173	46.7173	46.7173	46.7173	46.7173	4.67
10	47.9189	47.9189	47.9189	47.9189	47.9189	
15	46.7173	47.9189	47.5403	46.7173	47.9189	
20	46.7173	46.7173	46.7173	46.7173	46.7173	
25	49.2083	49.2083	49.2083	49.2083	49.2083	
30	54.6739	54.6739	54.6739	54.6739	54.6739	

Table 4.12 shows the thermal storage heat capacity for different output of biomass, gas boiler and solar collector at a fixed 15 kW heat pump. As the heat capacity of the heat pump increases, the upper limit of the biomass boiler heat capacity is reduced to 30 kW. It is shown that not all biomass boiler and gas boiler capacity can fulfil the peak heat demand. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was repeated for 15 kW heat pump as Table 4.13 shows.

Table 4.13 The reliability for all type of biomass boiler, heat boiler and solar collector at fixed 15 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	0.9842	0.9940	0.9994	1	1	1.56
10	0.9951	0.9988	0.9999	1	1	
15	0.9990	1	1	1	1	
20	09999	1	1	1	1	
25	1	1	1	1	1	
30	1	1	1	1	1	
5	0.9846	0.9949	0.9993	1	1	3.11
10	0.9954	0.9987	1	1	1	
15	0.9992	1	1	1	1	
20	0.9999	1	1	1	1	
25	1	1	1	1	1	
30	1	1	1	1	1	
5	0.9848	0.9947	0.9992	1	1	4.67
10	0.9950	0.9990	1	1	1	
15	0.9992	1	1	1	1	
20	1	1	1	1	1	
25	1	1	1	1	1	
30	1	1	1	1	1	

The results of the analysis of the thermal store capacities that can fulfil the heat demand of the site are further presented in the Table 4.14. The heat capacities of the thermal store that provides a reliability of energy supply less '1' have been discarded in this instance.

Table 4.14 The storage capacity for Scenario C with different heat generations

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	25	Solar collector (kW)
5	----------	----------	----------	29.5449	29.9905	1.56
10	----------	----------	----------	29.5449	29.6967	
15	----------	29.5449	29.4454	28.8026	29.8974	
20	----------	29.8516	29.8516	29.8516	29.8974	
25	34.5406	34.5406	34.5406	34.5406	34.5406	
30	38.9248	38.9248	38.9248	38.9248	38.9248	
5	----------	----------	----------	38.1743	38.1743	3.11
10	----------	----------	38.1743	38.1743	38.1743	
15	----------	39.2218	39.2218	39.2218	39.2218	
20	----------	38.1743	38.1743	38.1743	38.1743	
25	42.0642	42.0642	42.0642	42.0642	42.0642	
30	47.0642	47.0642	47.0642	47.0642	47.0642	
5	----------	----------	----------	46.7173	46.7173	4.67
10	----------	----------	47.9189	47.9189	47.9189	
15	----------	47.9189	47.5403	46.7173	47.9189	
20	46.7173	46.7173	46.7173	46.7173	46.7173	
25	49.2083	49.2083	49.2083	49.2083	49.2083	
30	54.6739	54.6739	54.6739	54.6739	54.6739	

Table 4.14 shows thermal store for reasonable biomass boilers and gas boilers with 3.1 m^2, 6.2 m^2, and 9.3 m^2 solar collector. It shows that increasing the solar thermal collector capacity has increased the storage capacity of the system without improvement in the flexibility of boiler rating required. The storage capacity with different heat sources at a fixed 15 kW heat pump can be seen in Figure 4.8.

Figure 4.8 shows the thermal store with different gas boiler, biomass boiler and solar collector at a fixed 15 kW heat pump. The lowest storage capacity with a 3.1 m^2 solar collector is 28.8 kWh, while the gas boiler and biomass boiler are rated at 20 kW and 15 kW respectively. For a solar collector of 6.2 m^2, the lowest storage capacity is 38.17 kWh with a lower gas boiler and biomass boiler rated at 15 kW and 10 kW respectively. Similarly, the lowest storage capacity of a 9.3 m^2 solar collector is 46.72 kWh, while gas boiler is reduced further to 5 kW and biomass boiler is 20 kW.

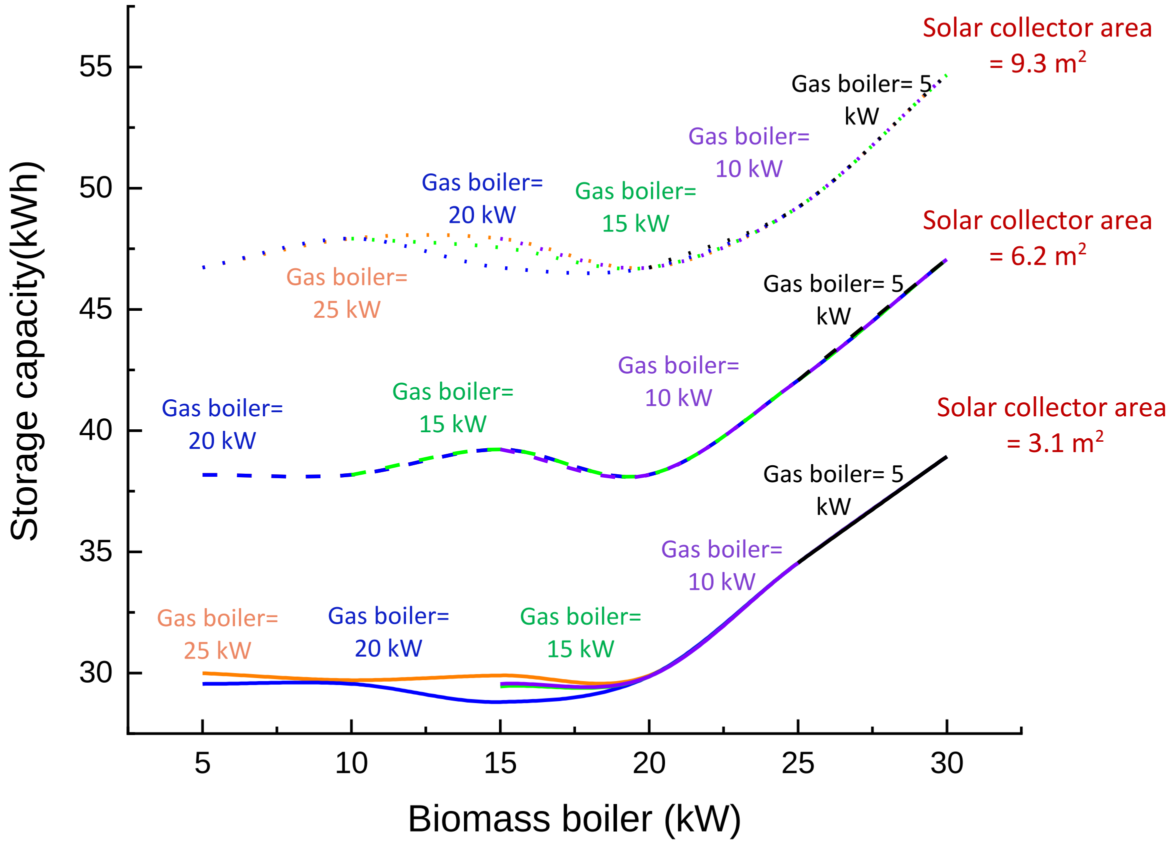


Figure 4.8 Scenario C: Storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 15 kW heat pump

4.4.5 Scenario D: 20 kW heat pump

Similar analytical procedure was repeated for Scenario D. The heat pump is fixed at 20 kW, and other heat generation outputs are allowed to vary. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW. The numerical simulation results for 1.56 kW, 3.11 kW and 4.67 kW solar collectors are tabulated in Table 4.15.

Table 4.15 The storage capacity for different output of biomass, gas boiler and solar collector at fixed 20 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	31.7977	31.7977	31.7977	31.7977	
10	31.7977	32.1465	31.7977	32.1465	
15	31.7977	31.7977	31.7977	31.7977	1.56
20	32.1465	32.1465	32.1465	32.1465	
25	34.8516	34.8516	34.8516	34.8516	
5	43.1249	43.1249	43.1249	43.1249	
10	43.1249	43.1249	43.1249	43.1249	
15	41.2319	41.2319	41.2319	41.2319	3.11
20	41.2319	41.2319	41.2319	41.2319	
25	43.1249	43.1249	43.1249	43.1249	
5	52.5403	52.5403	52.5403	52.5403	
10	52.5403	51.3455	52.5403	51.3455	
15	51.7173	51.7173	51.7173	51.7173	4.67
20	52.9189	52.9189	52.9189	52.9189	
25	52.5403	52.5403	52.5403	52.5403	

Table 4.15 shows the thermal storage heat capacity for different output of biomass, gas boiler and solar collector at a fixed 20 kW heat pump. As the capacity of the heat pump increases, the upper limit of the biomass boiler and gas boiler heat capacity is reduced to 25 kW and 20 kW respectively. It is shown that not all biomass boiler and gas boiler capacity can fulfil the peak heat demand. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was repeated for 20 kW heat pump as Table 4.16 shows.

Table 4.16 The reliability for all type of biomass boiler, heat boiler and solar collector at fixed 20 kW heat pump

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	Solar collector (kW)
5	0.9958	0.9990	1	1	
10	0.9991	0.9999	1	1	
15	0.9998	1	1	1	1.56
20	1	1	1	1	
25	1	1	1	1	
5	0.9962	0.9992	1	1	
10	0.9991	1	1	1	
15	0.9999	1	1	1	3.11
20	1	1	1	1	
25	1	1	1	1	
5	0.9958	0.9990	1	1	
10	0.9994	1	1	1	
15	1	1	1	1	4.67
20	1	1	1	1	
25	1	1	1	1	

The results of the analysis of the thermal store capacities that can fulfil the heat demand of the site are further presented in the Table 4.17. The heat capacities of the thermal store that provides a reliability of energy supply less '1' have been discarded in this instance.

Table 4.17 The storage capacity for Scenario D with different heat generations

Biomass boiler (kW) \ Gas boiler (kW)	5	10	15	20	Solar collector (kW)
5	----------	----------	31.7977	31.7977	
10	----------	----------	31.7977	32.1465	
15	----------	31.7977	31.7977	31.7977	1.56
20	32.1465	32.1465	32.1465	32.1465	
25	34.8516	34.8516	34.8516	34.8516	
5	----------	----------	43.1249	43.1249	
10	----------	43.1249	43.1249	43.1249	
15	----------	41.2319	41.2319	41.2319	3.11
20	41.2319	41.2319	41.2319	41.2319	
25	43.1249	43.1249	43.1249	43.1249	
5	----------	----------	52.5403	52.5403	
10	----------	51.3455	52.5403	51.3455	
15	51.7173	51.7173	51.7173	51.7173	4.67
20	52.9189	52.9189	52.9189	52.9189	
25	52.5403	52.5403	52.5403	52.5403	

Table 4.17 shows storage capacity for reasonable biomass boilers and gas boilers with 3.1 m^2, 6.2 m^2 and 9.3 m^2 solar collector. It shows that increasing the solar thermal collector capacity has increased the storage capacity of the system without improvement in the flexibility of boiler rating required. The storage capacity with different heat sources at a fixed 20 kW heat pump can be seen in Figure 4.9.

Figure 4.9 shows the storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 20 kW heat pump. The lowest storage capacity with a 3.1 m^2 solar collector is 31.8 kWh, while the gas boiler and biomass boiler are rated at 10 kW and 15 kW respectively. For a solar collector of 6.2 m^2, the lowest storage capacity is 41.23 kWh with a lower gas boiler and biomass boiler rating of 5 kW and 20 kW respectively. Similarly, the lowest storage capacity of a 9.3 m^2 solar collector is 51.35 kWh, while gas boiler and biomass boiler heating capacities are reduced further to 10 kW each.

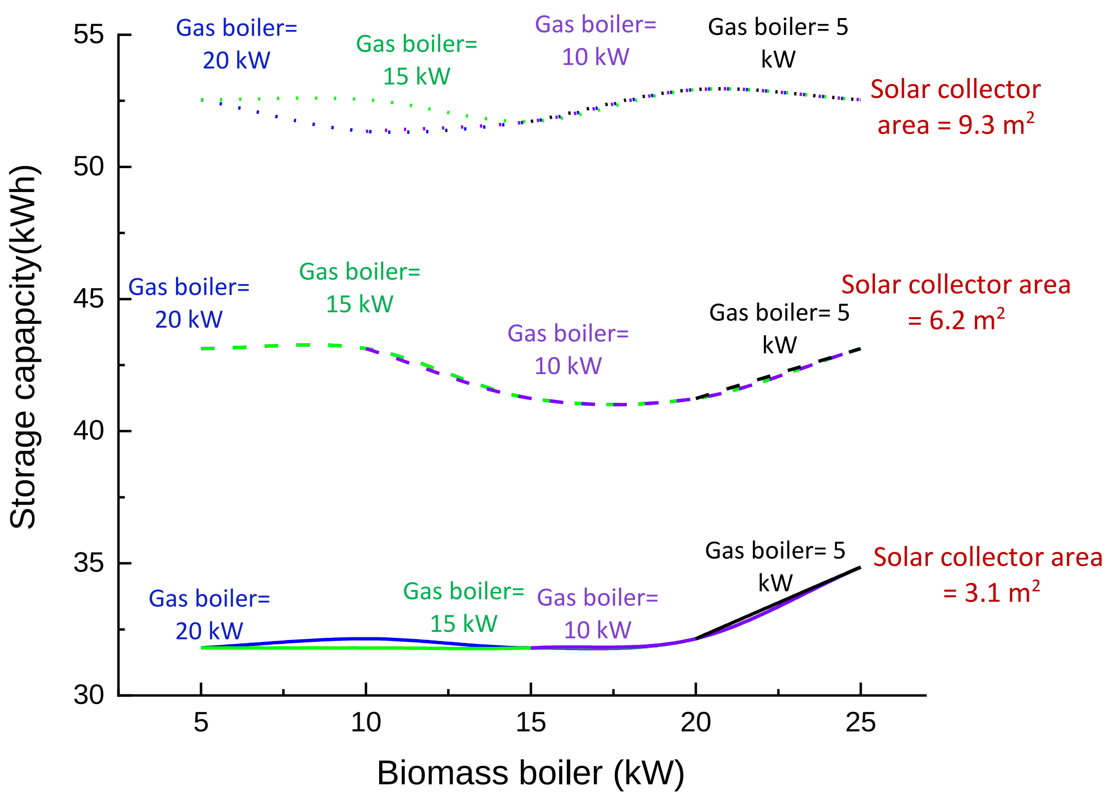


Figure 4.9 Scenario D: Storage capacity with different gas boiler, biomass boiler and solar collector at fixed 20 kW heat pump.

4.4.6 Scenario E: 25 kW heat pump

Similar analytical procedure was repeated for Scenario E. The heat pump is fixed at 25 kW, other heat generation outputs are allowed to vary. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW respectively. The numerical simulation results for 1.56 kW, 3.11 kW and 4.67 kW solar collector are tabulated in Table 4.18.

Table 4.18 The storage capacity for different output of biomass, gas boiler and solar collector at fixed 25 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	39.4454	39.4454	39.4454	39.4454	1.56
10	37.1465	37.1465	37.1465	37.1465	
15	39.4454	39.4454	39.4454	39.4454	
20	37.6198	37.6198	37.6198	37.6198	
5	48.1249	48.1249	48.1249	48.1249	3.11
10	48.1249	48.1249	48.1249	48.1249	
15	48.4929	48.4929	48.4929	48.4929	
20	49.2218	49.2218	49.2218	49.2218	
5	52.9189	52.9189	52.9189	52.9189	4.67
10	52.9189	52.9189	52.9189	52.9189	
15	56.7173	56.7173	56.7173	56.7173	
20	52.9189	52.9189	52.9189	52.9189	

Table 4.18 shows the thermal storage heat capacity for different output of biomass, gas boiler and solar collector at a fixed 25 kW heat pump. As the capacity of the heat pump increases, the upper limit of the biomass boiler heat capacity is further reduced to 20 kW. It is shown that not all biomass boiler and gas boiler capacity can fulfil the peak heat demand. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was repeated for 25 kW heat pump as Table 4.19 shows.

Table 4.19 The reliability for all type of biomass boiler, heat boiler and solar collector at fixed 25 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	0.9995	0.9999	1	1	1.56
10	0.9999	1	1	1	
15	1	1	1	1	
20	1	1	1	1	
5	0.9994	0.9999	1	1	3.11
10	1	1	1	1	
15	1	1	1	1	
20	1	1	1	1	
5	0.9997	1	1	1	4.67
10	1	1	1	1	
15	1	1	1	1	
20	1	1	1	1	

The results of the analysis of the thermal store capacities that can fulfil the heat demand of the site are further presented in the Table 4.20. The heat capacities of the thermal store that provides a reliability of energy supply less '1' have been discarded in this instance

Table 4.20 The storage capacity for Scenario E with different heat generations

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	----------	----------	39.4454	39.4454	1.56
10	----------	37.1465	37.1465	37.1465	
15	39.4454	39.4454	39.4454	39.4454	
20	37.6198	37.6198	37.6198	37.6198	
5	----------	----------	48.1249	48.1249	3.11
10	48.1249	48.1249	48.1249	48.1249	
15	48.4929	48.4929	48.4929	48.4929	
20	49.2218	49.2218	49.2218	49.2218	
5	----------	52.9189	52.9189	52.9189	4.67
10	52.9189	52.9189	52.9189	52.9189	
15	56.7173	56.7173	56.7173	56.7173	
20	52.9189	52.9189	52.9189	52.9189	

Table 4.20 shows storage capacity for reasonable biomass boilers and gas boilers with 3.1 m^2, 6.2 m^2 and 9.3 m^2 solar collector. It shows that increasing the solar thermal collector capacity has increased the storage capacity of the system without improvement in

the flexibility of boiler rating required. The storage capacity with different heat sources at a fixed 25 kW heat pump can be seen in Figure 4.10.

Figure 4.10 shows the storage capacity with different gas boiler, biomass boiler, solar collector at a fixed 25 kW heat pump. The lowest storage capacity with a 3.1 m^2 solar collector is 37.15 kWh, while both gas boiler and biomass boiler are rated at 10 kW. For a solar collector of 6.2 m^2, the lowest storage capacity is 48.13 kWh with a lower gas boiler and biomass boiler rating of 5 kW and 10 kW respectively. Similarly, the lowest storage capacity of a 9.3 m^2 solar collector is 52.92 kWh, while gas boiler and biomass boiler are rated at 5 kW and 10 kW respectively.

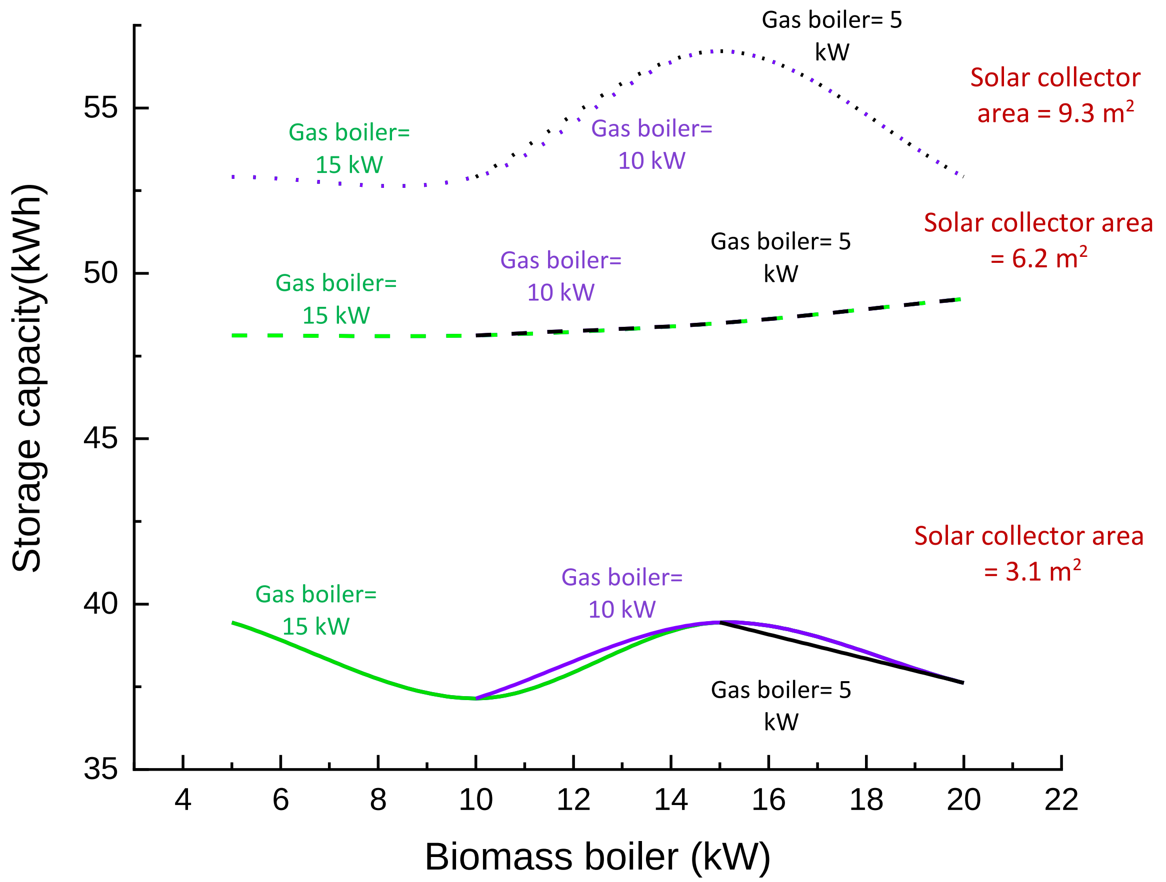


Figure 4.10 Scenario E: Storage capacity with different gas boiler, biomass boiler and solar collector at fixed a 25 kW heat pump

4.4.7 Scenario F: 30 kW heat pump

The heat pump is fixed at 30 kW, and other heat generation outputs are allowed to vary. Similar analytical procedure was repeated for Scenario F. In this scenario, the solar collector output is set to 1.56 kW, 3.11 kW and 4.67 kW respectively. The numerical simulation results for 1.56 kW, 3.11 kW and 4.67 kW solar collectors are tabulated in Table 4.21.

Table 4.21 The storage capacity for different output of biomass, gas boiler and solar collector at fixed 25 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	41.2249	41.2249	41.2249	41.2249	1.56
10	42.6198	42.6198	42.6198	42.6198	
15	44.4454	44.4454	44.4454	44.4454	
20	43.8026	43.8026	43.8026	43.8026	
5	51.2319	51.2319	51.2319	51.2319	3.11
10	49.0716	49.0716	49.0716	49.0716	
15	51.2319	51.2319	51.2319	51.2319	
20	51.2319	51.2319	51.2319	51.2319	
5	62.5403	62.5403	62.5403	62.5403	4.67
10	61.3455	61.3455	61.3455	61.3455	
15	62.5403	62.5403	62.5403	62.5403	
20	61.3455	61.3455	61.3455	61.3455	

Table 4.21 shows the thermal storage heat capacity for different output of biomass, gas boiler and solar collector at a fixed 30 kW heat pump. To illustrate the combination of the technologies that can satisfy the hourly peak load throughout the year, a reliability analysis was repeated for 30 kW heat pump as Table 4.22 shows. All the reliability for different generations is ‘1’, which means different sizes of heat generation can fulfil the heat load throughout the year. The analysis results of the thermal store capacities that can fulfil the heat demand of the site are shown in the Table 4.21.

Table 4.22 The reliability for all type of biomass boiler, heat boiler and solar collector at fixed 25 kW heat pump

Gas boiler (kW) / Biomass boiler (kW)	5	10	15	20	Solar collector (kW)
5	1	1	1	1	1.56
10	1	1	1	1	
15	1	1	1	1	
20	1	1	1	1	
5	1	1	1	1	3.11
10	1	1	1	1	
15	1	1	1	1	
20	1	1	1	1	
5	1	1	1	1	4.67
10	1	1	1	1	
15	1	1	1	1	
20	1	1	1	1	

The storage capacity with different heat sources at a fixed 25 kW heat pump can be seen in Figure 4.11 which shows the storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 30 kW heat pump. The storage capacity profile with a 3.1 m^2 solar collector is same no matter how biomass boiler and gas boiler heat capacities change as well as 6.2 m^2 solar collector and 9.2 m^2 solar collector.

The lowest storage capacity with a 3.1 m^2 solar collector is 41.23 kWh, while both gas boiler and biomass boiler are rated at 5 kW. For a solar collector of 6.2 m^2, the lowest storage capacity is 49.07 kWh with the gas boiler and biomass boiler rated at 5 kW and 10 kW respectively. Similarly, the lowest storage capacity of a 9.3 m^2 solar collector is 61.34 kWh, while the gas boiler and biomass boiler rated at 5 kW and 10 kW respectively. As the heat pump is fixed at 30 kW, heat generation capacity no matter how these vary and thermal store can satisfy the hourly heat demand reliably throughout the year. There is no need of increasing the heat capacity of a gas boiler.

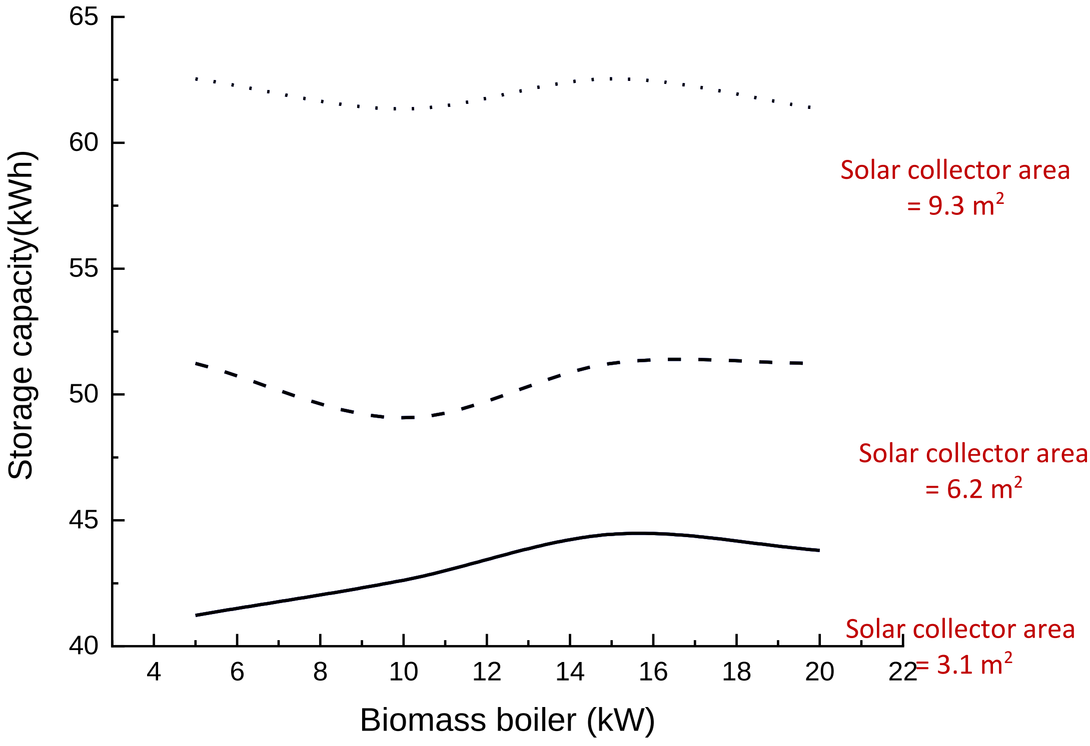


Figure 4.11 Scenario F: Storage capacity with different gas boiler, biomass boiler and solar collector at a fixed 30 kW heat pump

4.4.8 Dynamic analysis of thermal store capacity

The variation of the amount heat stored in the thermal store throughout the year in accordance with the site's heat demand profile was analysed. There are different heat generation outputs and the size of solar thermal collector, the heat pump, gas boiler and solar collector outputs are fixed, while the biomass boiler heat capacity is allowed to vary in order to dynamic analyse.

For example, the heat pump is fixed at 15 kW, while the gas boiler and solar collector rated 20 kW and 4.67 kW respectively. The biomass boiler output is allowed to vary. Figure 4.12 shows the variation of amount heat stored in the thermal store throughout the year with 5 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector. It is shown the maximum energy stored in the thermal stored during the year occurred in June, reaching 46.72 kWh.

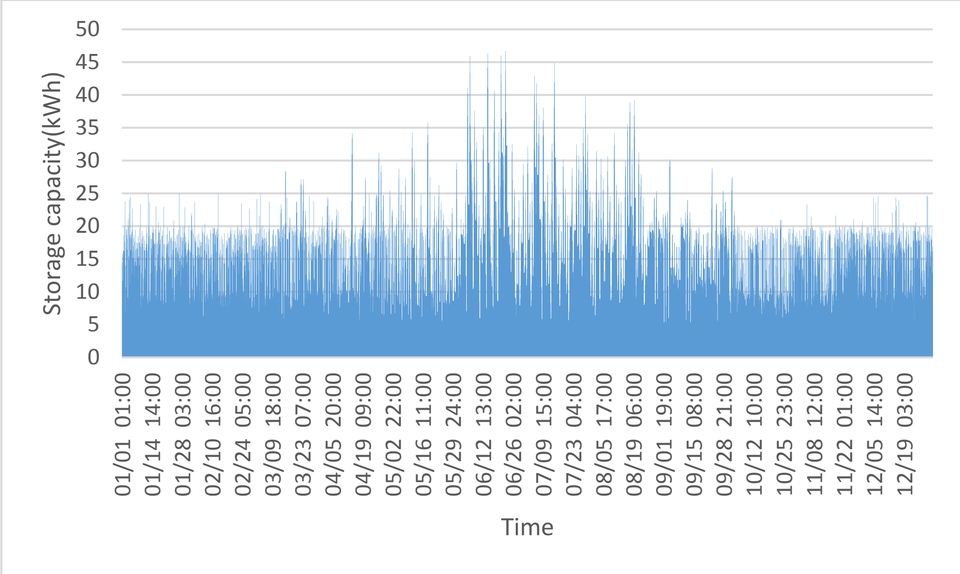


Figure 4.12 The variation of amount heat stored in the thermal store throughout the year with 5 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

The instantaneous variation of the amount of stored heat in relation to 5 kW biomass boiler during a representative week of month of June is selected as Figure 4.13 shows. For thermal store, the heat generations charge it, which mean that the heat is input into thermal store. As a result, the heat capacities of heat generations are considered as positive value. However, the heat demand is satisfied by the thermal store, which means the heat is output from thermal store to buildings. As a result, the heat demand is considered as negative value.

As Figure 4.13 shows, the maximum heat demand during a day always occurred at morning due to the heat demand and low temperature. The storage capacity was 19.59 kW at 06:00 on June 22nd, which was accumulated from previous and during this period. Then, the heat demand was 14.3 kW. The thermal store capacity was sufficient to supply the heat demand, after that the thermal store capacity was 5.29 kW. Although the heat demand in the following hour decreased to 3.07 kW, 0.45kW solar collector heat capacity in combination with the thermal store capacity cannot meet the heat demand. Because the residual thermal storage capacity cannot be less than minimum storage capacity, 15 kW heat pump switch on for one hour to supply the rest heat demand. As a result, the heat store capacity was increased to 17.9 kW. There is no heat demand in the next following hour and some solar energy continuously charged into the heat store. The heat pump switched off until the heat store cannot satisfy heat demand. The heat pump switched on six times and each time was one hour, mostly of them occurred in morning. The maximum storage capacity was 46.72 kW before 17:00 on June 22nd. The solar energy accumulated on the storage and the heat demand is 0 in afternoon. After 17:00, the heat demand became to increase, while the solar energy cannot satisfy it, the thermal store needed to discharge to supply the rest heat.

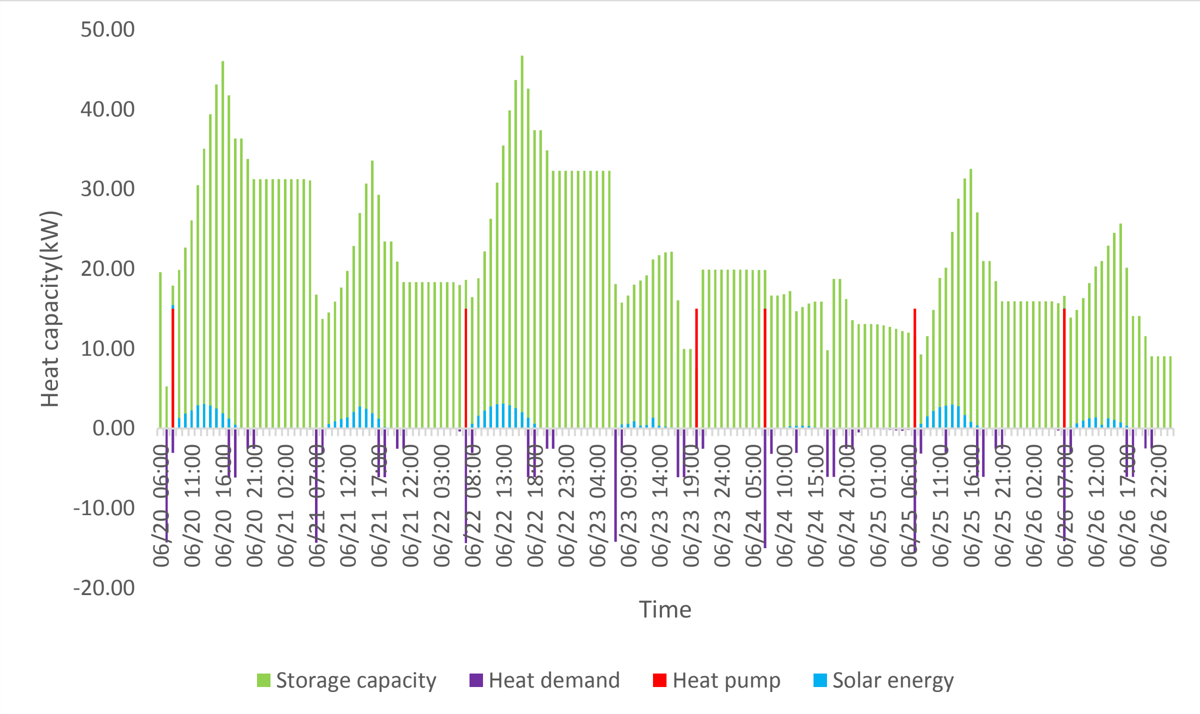


Figure 4.13 The variation of storage capacity with 5 kW biomass boiler during a representative week of month of June

The similar analytical procedure was repeated for heat output capacities of 10 kW, 15 kW and 20 kW biomass boilers, respectively. The results of amount variation of heat stored in the thermal store throughout the year can be seen in Figure 4.14, Figure 4.15 and Figure 4.16 respectively. Figure 4.14, Figure 4.15 and Figure 4.16 show the maximum energy stored in the thermal stored during the year occurred in July, June and June respectively.

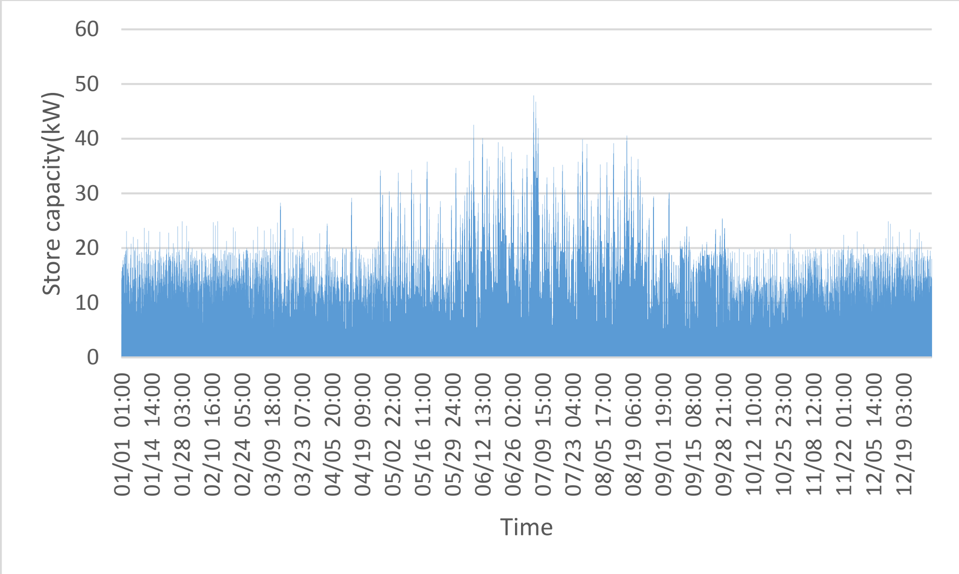


Figure 4.14 The variation of amount heat stored in the thermal store throughout the year with 10 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

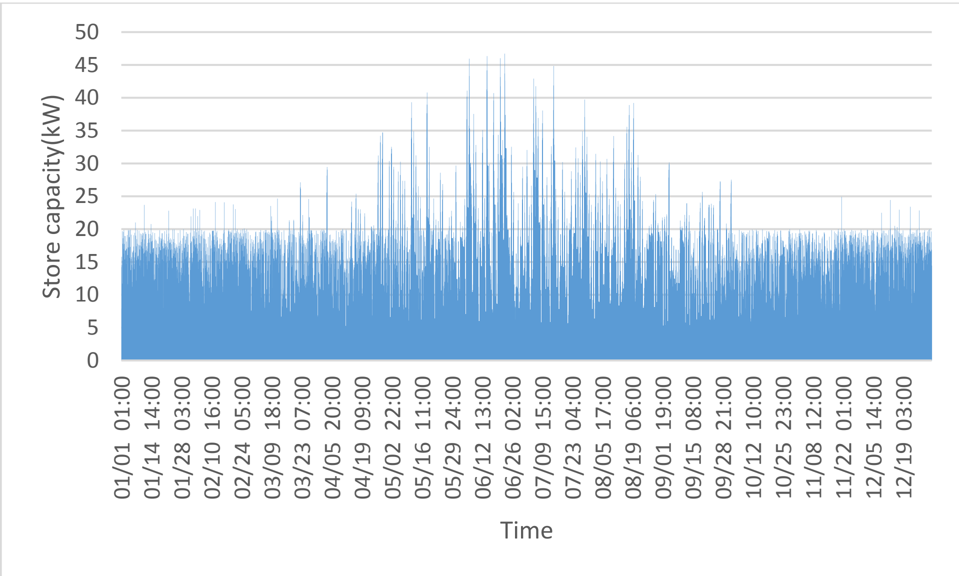


Figure 4.15 The variation of amount heat stored in the thermal store throughout the year with 15 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

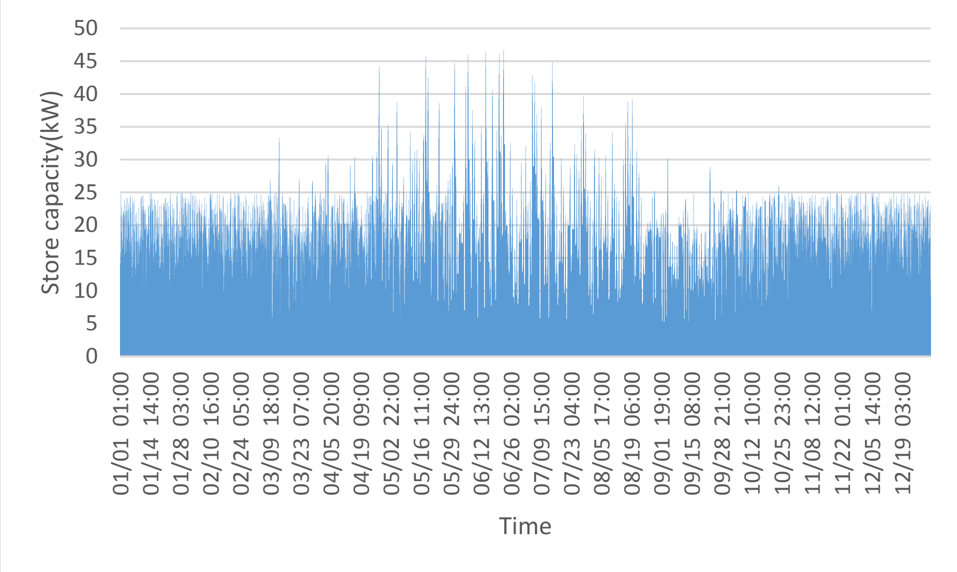


Figure 4.16 The variation of amount heat stored in the thermal store throughout the year with 20 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

The instantaneous variations in the amount of stored heat in relation to 10 kW, 15 kW and 20 kW biomass boilers during a representative week of the month are selected as Figure 4.17, Figure 4.18 and Figure 4.19 show respectively.

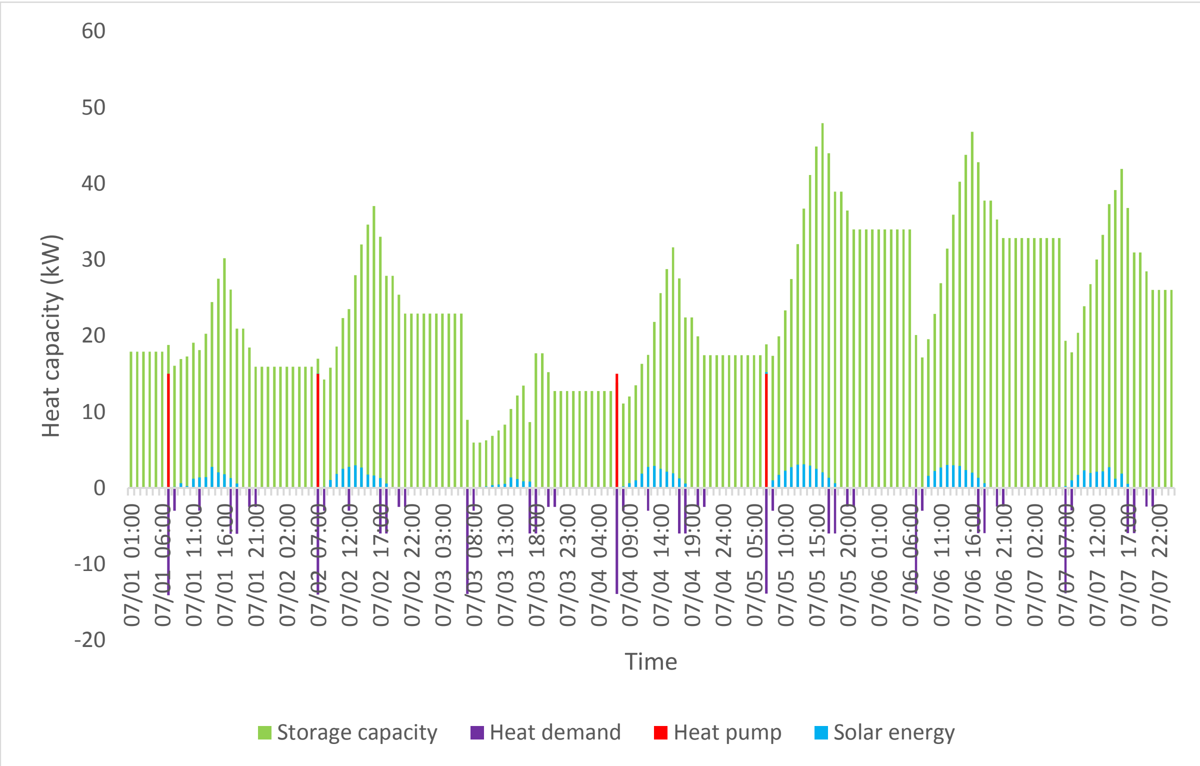


Figure 4.17 The variation of storage capacity with 10 kW biomass boiler during a representative week of month of July

Figure 4.17 shows that the maximum storage capacity with a 10 kW biomass boiler was 47.92 kW, which appeared before 17:00 on July 5th. The heat pump switched on 4 times and ran for one hour each time, all of which occurred in morning.

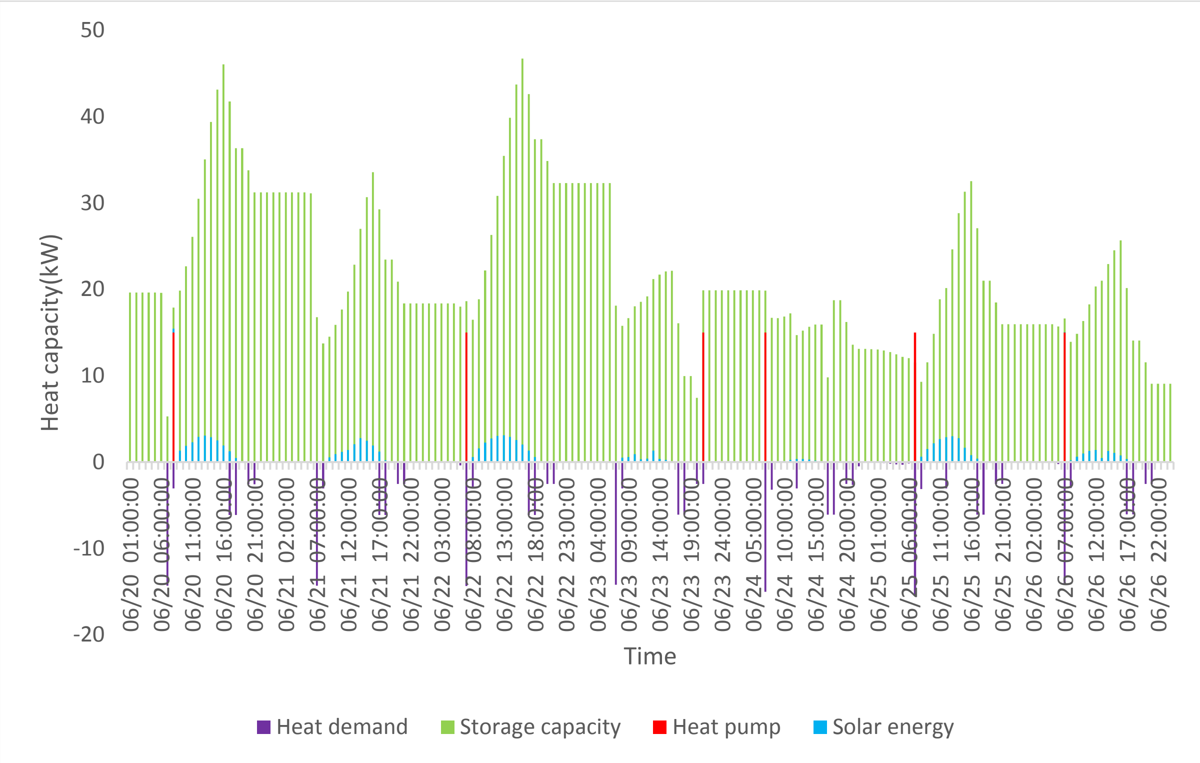


Figure 4.18 The variation of storage capacity with 15kW biomass boiler during a representative week of month of June

As Figure 4.18 and Figure 4.19 show, the variation of storage capacity with 15 kW and 20 kW biomass boilers during a representative week of the month of June are same. The maximum storage capacity for them was 46.72 kW, which appeared before 17:00 on June 22nd. The heat pump switched on 5 times and worked for one hour each time, 80% of which occurred in morning.

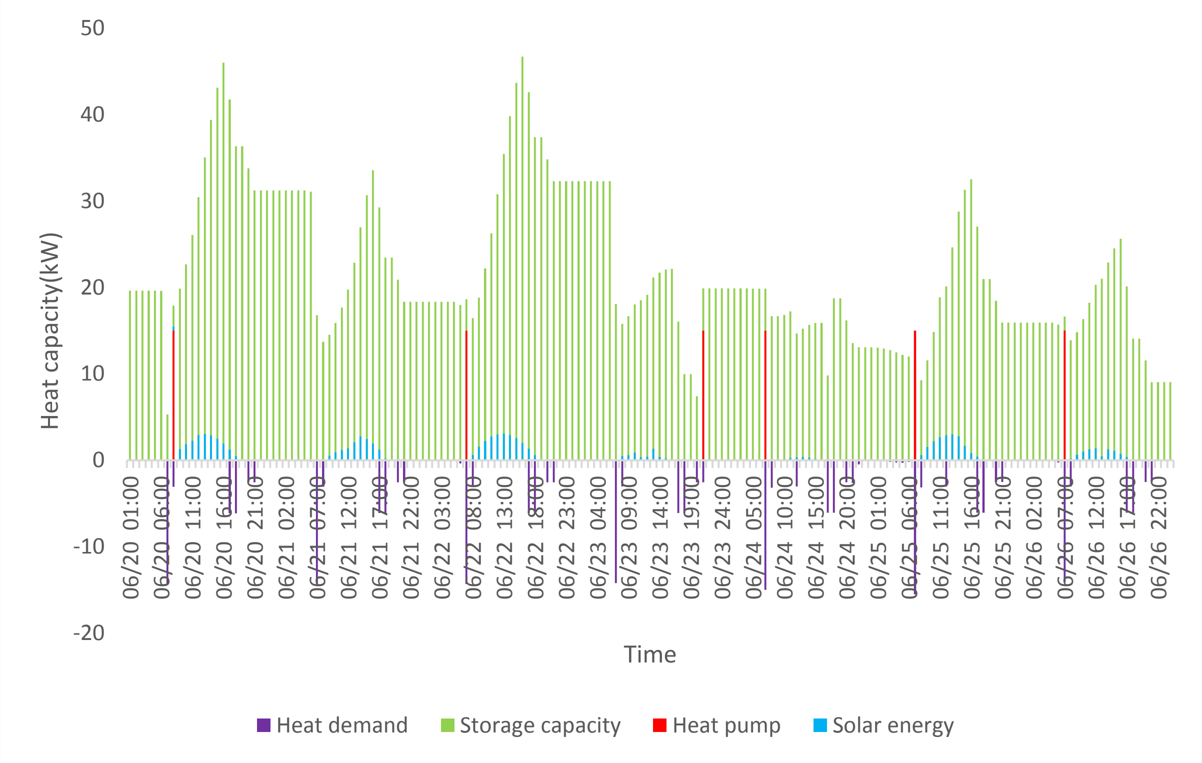


Figure 4.19 The variation of storage capacity with 20 kW biomass boiler during a representative week of month of June

The similar analytical procedure was repeated for heat capacities of 25 kW and 30 kW biomass boilers, respectively. The results of amount variation of heat stored in the thermal store throughout the year can be seen in Figure 4.20 and Figure 4.21 respectively. Figure 4.20 and Figure 4.21 show the maximum energy stored in the thermal stored during the year occurred in April and May respectively, which are not in summer (June and July). As a result, the amount variation of heat stored during a representative week of the month is different due to the priority of heat generations.

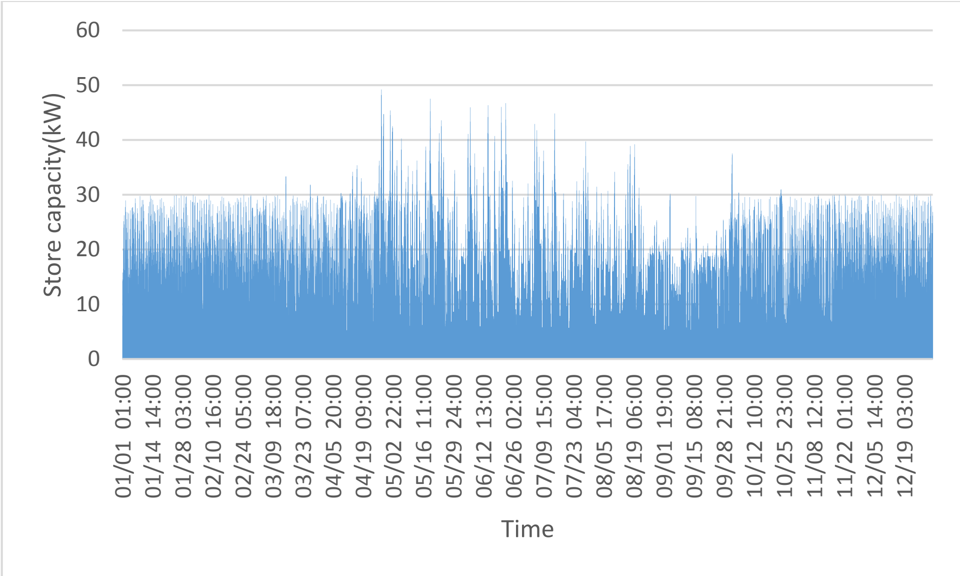


Figure 4.20 The variation of amount heat stored in the thermal store throughout the year with 25 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

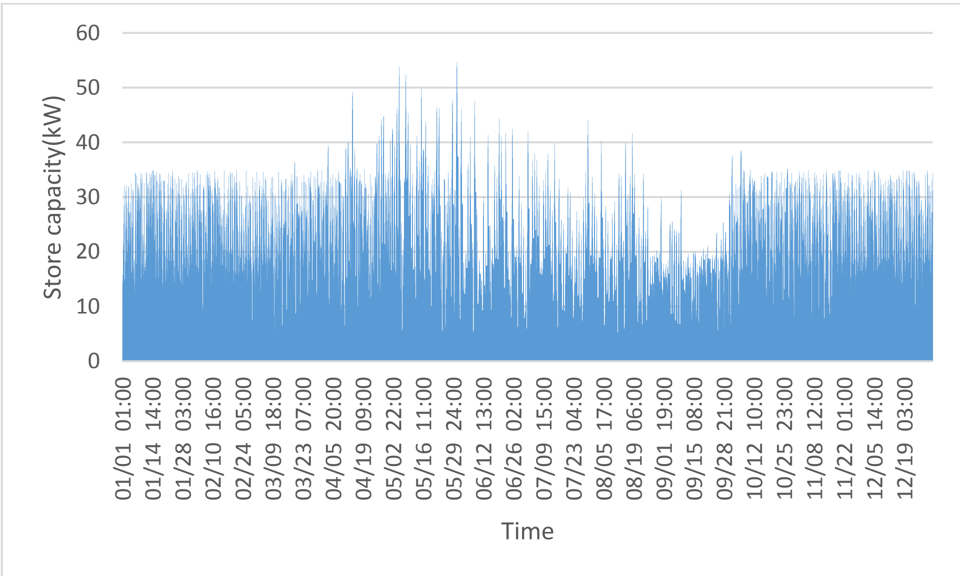


Figure 4.21 The variation of amount heat stored in the thermal store throughout the year with 30 kW biomass boiler, 15 kW heat pump, 20 kW gas boiler and 4.16 kW solar collector

For example, the instantaneous variation of the amount of heat stored in relation to 25 kW biomass boiler during a representative week of month of April is selected as Figure 4.22 shows. The thermal store capacity was 6.48 kW at 01:00 which was accumulated from previous day, the thermal store can satisfy the next two hours' heat load without other heat generation until it reached 5.56 kW. The residual thermal storage capacity was not sufficient to meet heat demand, while 15 kW heat pump switched on for one hour according to priority order. Then, it switched off until 07:00, while the heat demand was 21.63 kW, the residual thermal storage capacity cannot satisfy it. The heat pump switched on for one hour, however, the sum of heat pump output and thermal store capacity cannot satisfy the heat demand. As a result, 25 kW biomass boiler in conjunction with 15 kW heat pump worked for one hour to satisfy the heat demand, while surplus heat was accumulated in thermal store. After that, biomass boiler and heat pump switched off until to 21:00, while thermal store cannot satisfy the heat demand. Biomass boiler switched on for only one hour to supply the heat according to priority order. The maximum store capacity occurred before 17:00 on April 27th, which was 49.21 kW. The heat pump switched on 10 times for 11 hours, because heat pump worked for 2 hours one time. Most of heat pump worked in morning. Biomass boiler switched on 12 times and each time is only one hour, of which 3 times biomass boiler worked in combination with heat pump.

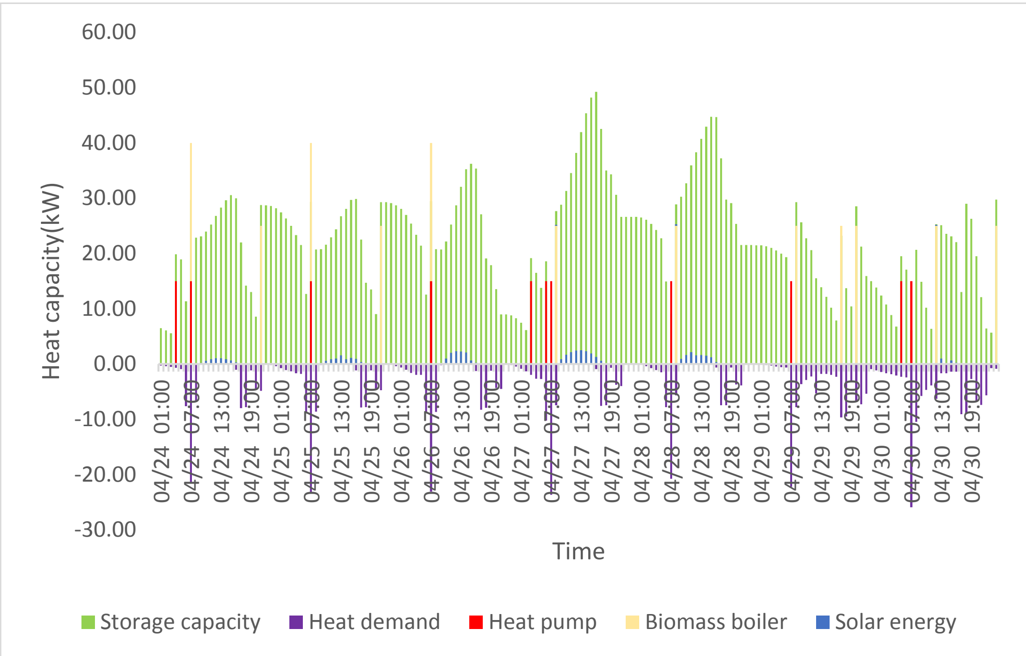


Figure 4.22 The variation of storage capacity with 25 kW biomass boiler during a representative week of month of April

Analysis of the Low Temperature Heat Network heat sources

The instantaneous variation in the amount of heat stored in relation to 30 kW biomass boiler during a representative week of the month of May is selected as Figure 4.23 shows. Similarly, the maximum store capacity occurred before 17:00 on May 31st, which was 54.67 kW with lower start up times of 15 kW heat pump and 30 kW biomass boiler rated at 4 and 7 respectively. Both biomass boiler and gas boiler switched on for only one hour. All the heat pump occurred in morning.

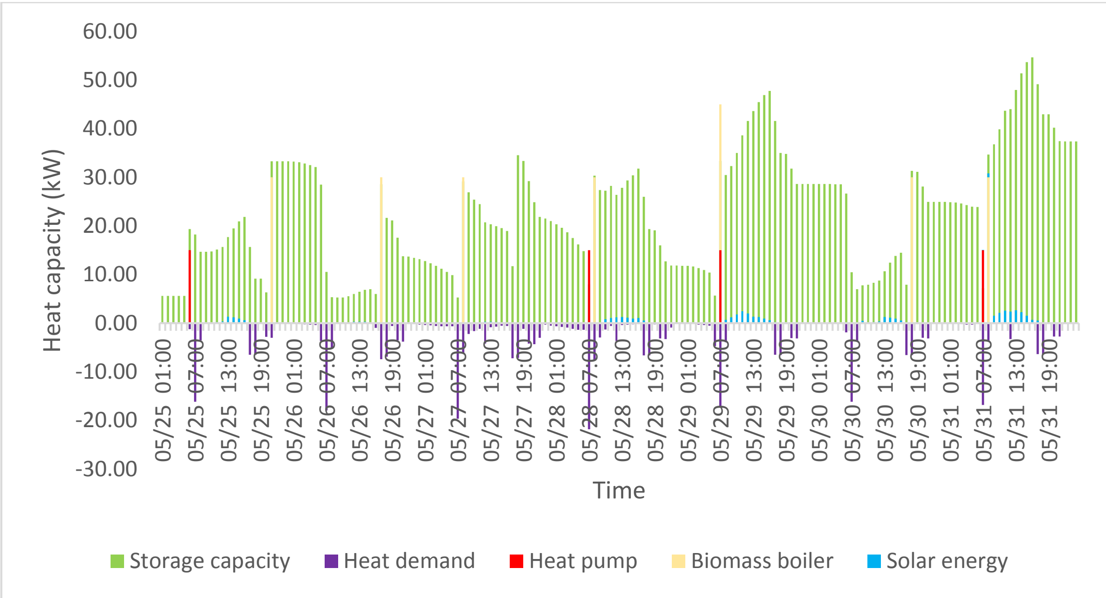


Figure 4.23 The variation of storage capacity with 30 kW biomass boiler during a representative week of month of May

Overall, dynamic analysis of thermal storage with different size biomass boiler have been studied. The maximum storage capacity with different size biomass boiler during a representative week always occurred before 17:00. For 5 kW to 20 kW biomass boiler, solar energy can charge the thermal store to satisfy mostly heat demand, the rest heat was met by heat pump. The heat pump worked as backup equipment mostly in morning. For 25 kW and 30kW biomass boiler, the representative week of month was in April and May respectively, the heat demand is higher than in July and June. As a result, the heat pump, solar collector and thermal store worked to satisfy the basic heat demand, while biomass boiler satisfied the peak load.

4.4.9 Optimisation of heat sources for different output solar collector

The lowest thermal store capacity for fixed heat pump (5 kW, 10 kW, 15 kW, 20 kW, 25 kW and 30 kW) with different size solar collector, biomass boiler and gas boiler has been obtained. It can be divided according to the three types of solar collectors.

Under the same output of a solar collector, the optimisation of heat generations abided by the following principles. Firstly, the smallest storage capacity is retained. Secondly, the sum heat output of biomass boiler, heat pump and gas boiler are the lowest when the storage capacity is the same. Lastly, clean energy system is selected when the storage capacity and output of heat generations are the same.

Table 4.23 Different heat generations with different thermal store capacities

Solar collector (kW)	Solar collector area (m^2)	Heat Pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Thermal storage (kW)
1.56	3.1	5	20	20	28.11
		10	15	20	26.68
		15	15	20	28.8
		20	15	10	31.8
		20	5	15	31.8
		25	10	10	37.15
		30	5	5	41.23
3.11	6.2	5	15	20	35.26
		10	10	20	33.12
		15	20	10	38.17
		15	10	15	38.17
		15	5	20	38.17
		20	20	5	41.23
		20	15	10	41.23
		25	10	5	48.13
		25	5	15	48.13
		30	10	5	49.07
4.67	9.3	5	10	25	44.21
		10	15	15	42.54
		15	20	5	46.72
		15	5	20	46.72
		20	10	10	51.35
		25	10	5	52.92
		25	5	10	52.92
		30	10	5	61.35

From Table 4.23, the lowest storage capacity to satisfy the site's energy demand is 26.68 kW which would require a combination of 10 kW heat pump, 15 kW biomass boiler and 20 kW gas boiler and for a solar collector is 1.56 kW (3.1 m^2). For 6.2 m^2 solar collector, the optimum heat sources are 10 kW heat pump, 10kW biomass boiler and 20kW gas boiler, while the storage capacity is 33.12 kWh. For 9.3 m^2 solar collector, the optimum heat sources are 10 kW heat pump, 15 kW biomass boiler and 15 kW gas boiler, while the storage capacity is 42.54 kWh. The optimised heat generations outputs and storage capacities are summarised in Table 4.24 shows.

Table 4.24 Three cases for optimised heat generations

CASE	Max output of solar collector	Solar collector area (m^2)	Heat pump(kW)	Biomass boiler (kW)	Gas boiler (kW)	Thermal storage(kW)
1	1.56	3.1	10	15	20	26.68
2	3.11	6.2	10	10	20	33.12
3	4.67	9.3	10	15	15	42.54

To select the optimum thermal store capacity among the three cases summarised in Table 4.24, a further insight into the instantaneous variation of the thermal capacity during a period of high thermal demand is investigated. The variation for the heat generation and storage capacity are then considered for the biggest thermal storage of the year. It was established through the model that the biggest storage capacity for the 1.56 kW solar collector occurred on July 6th. The load and heat generation profiles are shown in Figure 4.24, where it can be seen that in the morning (00:00 to 07:00) the thermal store capacity was 8.65 kWh which was accumulated from previous and during this period there is no demand for heat. A call for heat (domestic hot water) spiked at 07:00 with a peak load of 13.87 kW. Because the energy available from the thermal store is not enough to meet the heat load. A 10 kW heat pump and a 15 kW biomass boiler were both required to generate heat in the set of priority order. As no modulation of the heat generation system was considered, a surplus of energy was generated which contributed to the increasing the energy stored in the store capacity.

The increase of thermal store's energy meant that the heat pump and biomass boiler were not required for the rest the day and the thermal store alone and the solar collector were sufficient to meet the demand for hot water. During the day, the solar collector increased the thermal store energy by about 10.84 kWh about to a peak of 26.68 kWh.

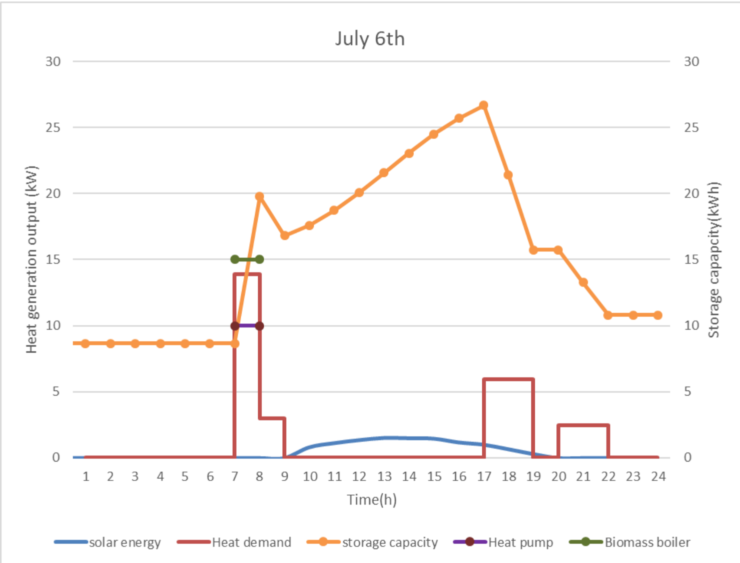


Figure 4.24 The heat generation and storage capacity variation for 1.56 kW solar collector on July 6th.

In a similar scenario, for the 6.2 m^2 solar collector (3.11 kW), the highest amount of energy stored in the thermal store was obtained on June 20th. The total initial amount of energy of the thermal store carried over from the previous day stood at 11.29 kWh before 07:00 as shown in Figure 4.25. The heat demand between 07:00 and 08:00 was modelled as a step increase of 14.3 kW. To supply the heat load for hot water, the 10 kW heat pump in combination with thermal store capacity was sufficient to supply the load for one hour. Even though the heat demand decreased in the following hour to 3.07 kW, the heat capacity of the thermal store alone was not sufficient to meet the heat demand as it was discharged to 6.99 kWh which requires the heat pump to operate for another hour in conjunction with the solar collector and recharge the thermal store in turn. By the next call for heat at 17:00, the amount of energy accumulated in the thermal store supplied by the solar collector reached 33.12 kWh.

For the remainder of the day the heat demand was all supplied by the thermal store with excess capacity carried over to the following day.

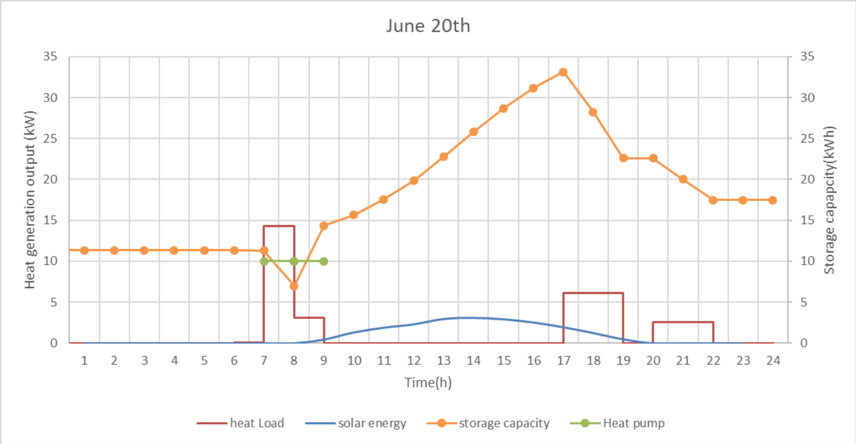


Figure 4.25 The heat generation and storage capacity variation for 3.11 kW solar collector on June 20th

The previous mode of the operation of the heat networks was replicated for the 9.3 m^2 (4.67 kW) solar collector as shown in Figure 4.26, the exception in that case is that the peak amount of heat stored in the thermal store increased to 42.54 kWh.

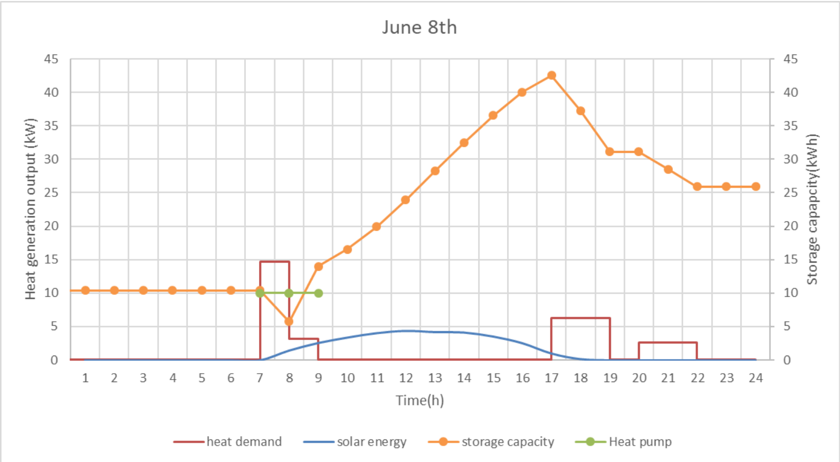


Figure 4.26 The heat generation and storage capacity variation for 4.67 kW solar collector on June 8th

From the previous dynamic analysis, the optimum storage capacity of the thermal store and the operation schedule of different generation systems was obtained.

Analysis of the Low Temperature Heat Network heat sources

To assess the contribution of each technology to the generation, Figure 4.27 shows the total number of working hours per year of each heat source for the case of a solar collector with 3.1 m^2. It can be seen that as the solar collector is taken as priority '1' for operation, it recorded the longest working hour of 1891 hours, while the gas boiler which is ranked lowest in the operation priority was only called upon for 85 hours. Ranked second in the total number of working hours is the biomass boiler with 1746 hours while the heat pump operated for a total of 1146 hours per year. In the absence of other heat generations for charging, the thermal energy storage can work for up to 24 hours.

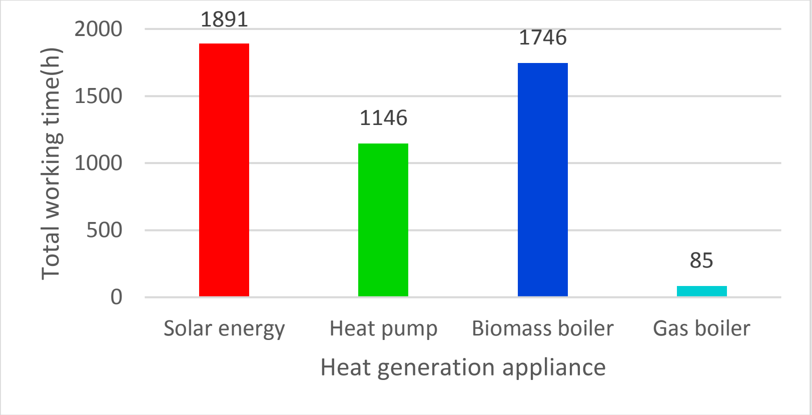


Figure 4.27 Total working time of each heat source in case 1.

The total number of working hours per year of each heat source for the case of a solar collector with 6.2 m^2 as Figure 4.28 shows. It can be seen that the solar collector recorded with same hours of 1891 hours, while it ranked second in the total number of working hours. However, the working hours of biomass boiler increased to the longest working hours of 2042 hours, which recorded the first of total number of working hours. Similarly, gas boiler ranked lowest in the operation priority with higher working hours for 188 hours, while heat pump operated for a total of 1425 hours per year. In the absence of other heat generations for charging, the thermal energy storage can work for up to 25 hours.

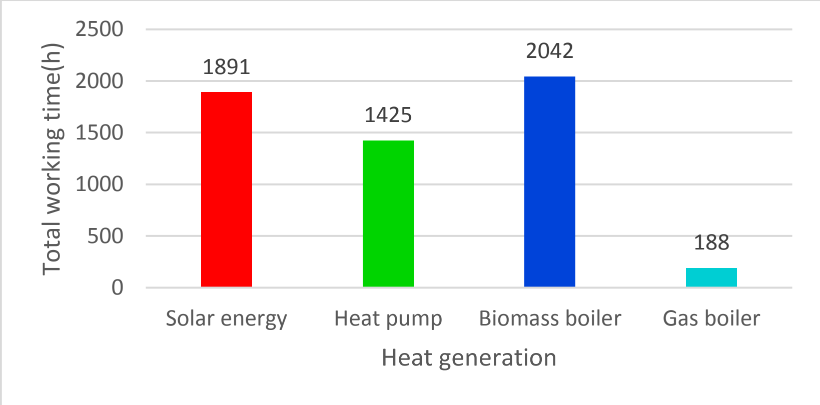


Figure 4.28 Working time of each heat source in case 2.

As Figure 4.29 shows, the previous ranking of working hours for heat generation in case 1 was replicated for the 9.3 m^2 solar collector, the exception in that case 3 was that number of working hours for heat pump, biomass boiler and gas boiler, which was 1043 hours, 1724 hours and 82 hours respectively. The longest working hour of 1891 hours was solar collector. In the absence of other heat generations for charging, the thermal energy storage can work for up to 96 hours in summer due to the increase of solar collector area. As it rains 1 in 3 days in England on average, the solar energy is little in winter which has little impact on solar collector. In summer, heat pump can work while the solar collector is not functioning in summer.

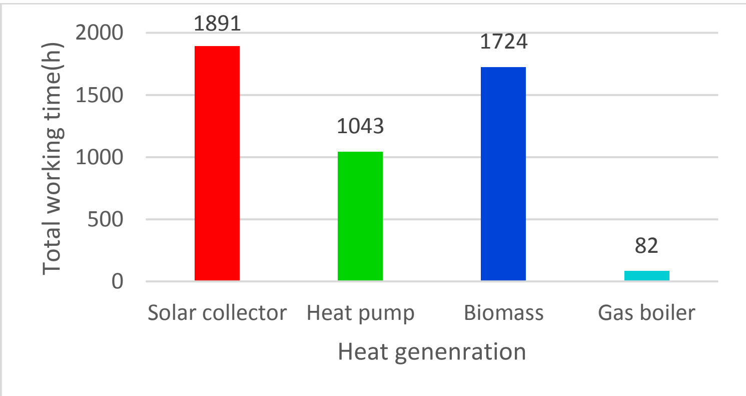
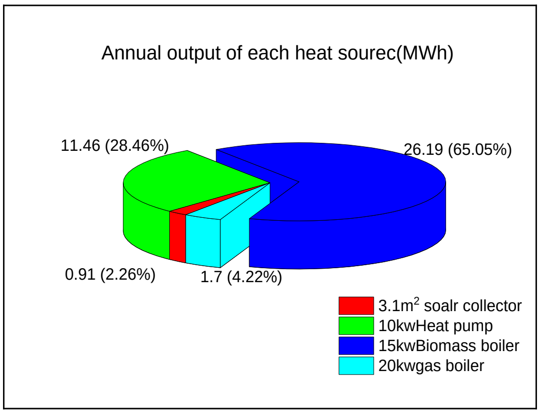
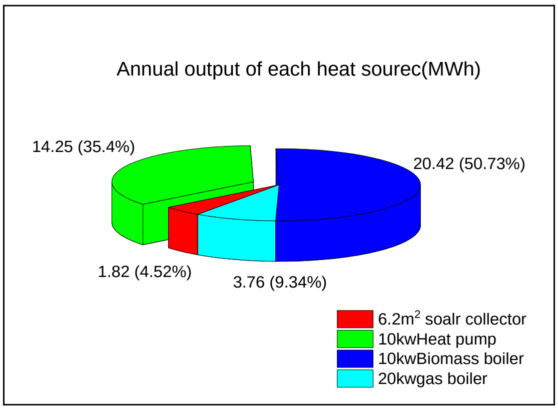


Figure 4.29 Working time of each heat source in case 3

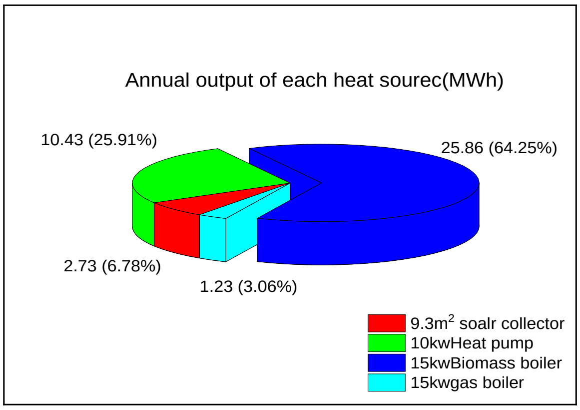
4.4.10 Annual thermal energy contribution mix

The annual energy generation contribution of each heat source for the three cases is shown in Figure 4.30 (a, b, c). The largest annual output was biomass boiler for three cases, which was 26.19 MWh, 20.45 MWh, 25.86 MWh respectively. The biomass boiler for three cases accounted for more than 50% of the total output of heat sources. The heat pump was the second largest annual output for three cases, representing 11.46 MWh, 14.25 MWh, 10.43 MWh respectively. The third largest one for case 1 and case 2 was the gas boiler, while the last one is solar collector. However, the third largest contrition device was solar collector in case 3, while the last one was gas boiler, which was led by the increase of solar collector heat capacity.





a) Solar collector 3.1 m^2 b) Solar collector 6.2 m^2



c) Solar collector 9.3 m^2

Figure 4.30 Annual thermal energy contribution by heat source

4.5 Summary

The developed methodology was used to optimise the heat sources and obtain the heat storage capacity through annual heat demand, different output of heat generations (solar collector, heat pump, biomass boiler and gas boiler) and reliability analysis. The reasonable different sizes of heat generation can be obtained from Scenario A-F.

Dynamic analysis of thermal storage capacity was established for different size biomass boiler during a representative week of month. The biggest storage capacity during a year always occurred before 17:00, because solar energy continuously charged thermal store and the heat demand is 0 from 9:00-17:00. The variation of storage capacity profile was caused by biggest thermal capacity's month, the operation priority of heat sources was variable in different months. Solar energy was the basic heat source to supply the heat load, while the heat pump was as an auxiliary heat source to supply the rest energy which cannot be supplied by the thermal store and solar energy during July and June. The basic heat source was heat pump and thermal store, biomass boiler works as a backup heat source during April and May. In addition, there is a little solar energy to charge the thermal store.

Optimisation for the heat source for different solar collector can be divided into three cases.

Case1: 1.56 kW solar collector, 10 kW heat pump 15 kW biomass boiler and 20 kW gas boiler with 26.68 kWh thermal store capacity.

Case2: 3.11 kW solar collector, 10 kW heat pump 10 kW biomass boiler and 20 kW gas boiler with 33.12 kWh thermal store capacity.

Case 3: 4.67 kW solar collector, 10 kW heat pump 15 kW biomass boiler and 15 kW gas boiler with 42.54 kWh thermal store capacity

The operation of heat sources and storage variation on the day of maximum storage capacity for three cases. The working time of solar collectors for three cases were all 1891 hours, ranking 1st, 2nd, 1st in length of time. Excluding the solar collector working time, remaining heat sources were sorted by total working hours. The longest one was biomass boiler, the next one was heat pump and the last one was gas boiler. In the absence of other heat generations for charging, the thermal energy storage can work for up 4 days in summer. As it rains 1 in 3 days in England on average, the solar energy is little in winter which has little impact on solar collector. In summer, heat pump can work while the solar collector is not functioning in summer.

Although the working time of solar collector is relatively long, the solar collector is generally ranked in the last two for three cases according to the total output of each heat source due to to the global horizontal radiation. The largest generation for three cases was all biomass boilers, while the heat pump was the second largest heat generation. Biomass boiler and heat pump contributed more than 85% of heat demand, which were the main heat generations for three cases. Gas boiler covered less than 10% of heat demand, working as auxiliary boiler.

5 Optimisation of the thermal store

5.1 Introduction

In this chapter, the thermal storage including heat loss is optimised for integration into LTDH system. A MATLAB program was developed to solve a dynamic mathematical model to simulate the hourly variation of heat demand of the site. The simulation also includes optimised heat generation form the heat sources.

Firstly, the mathematical model for LTDH system with heat loss of thermal storage is established in section 5.2. Secondly, the optimisation of thermal store capacity was considered for three scenarios as shown in section 5.3. The overall effect of heat loss of the thermal store was also assessed in section 5.4. The chapter findings are finally summarised in section 5.5.

5.2 Methodology

The thermal store capacity is established from the heat balance of heat generations from all contributing heat sources and heat demand of the site on hourly basis. To improve the accuracy of the model, the heat loss from the thermal store is taken into account in this.

The thermal capacity of the thermal store is usually specified by the industry by the water content volume which in turn can established the physical size of the component. Therefore, the volume of the thermal store can be evaluated by the following equation:

$$V = \frac{Q_{s-big} * 3.6 * 10^6}{C_P \rho \Delta T_s} \qquad (5-1)$$

Where V is the volume of thermal store (L), Q_{s-big} is the maximum of energy stored in thermal store (kWh), C_P is the specific heat capacity of water $(kJ/(kg \cdot K))$, ρ is the density of water (kg/m^3), ΔT_s is the temperature difference for thermal store supply/ return temperature (°C).

Similarly, the heat loss of the thermal store through its outer walls depends on the level of thermal insulation and ambient temperature. This can be expressed by the following equation

$$Q^{loss} = UA\Delta T/1000 \qquad (5-2)$$

Where Q^{loss} is heat loss of thermal store (kW), U is overall heat transfer coefficient ($W/(m^2K)$), A is the surface area of thermal store (m^2), ΔT is the temperature difference between the water temperature and ambient temperature (°C). The surface area of a cylindrical shape thermal store is determined by

$$A = \pi dh + \frac{1}{2}\pi d^2 \qquad (5-3)$$

Where d is the diameter of thermal store (m), h is the height of thermal store (m).

Combined with Equation (5-2) and Equation (5-3), the heat loss can be expressed

$$Q^{loss} = U \cdot \left(\pi dh + \frac{1}{2}\pi d^2\right) \cdot (T_{in} - T_a)/1000 \qquad (5-4)$$

Where T_{in} is the average temperature of water inside the thermal store (℃), T_a is the ambient temperature (℃). The heat transfer coefficient is assumed same for horizontal and vertical faces.

By taking into account heat loss from the thermal store, its energy balance developed in Equation (4-3) can be re-written as follows:

$$Q^{Storage} = Q^{Intial} + Q^S + Q^H + Q^B + Q^G - Q^{Out} - Q^{loss} \qquad (5-5)$$

The thermal store energy balance of Equation (5-5) can be applied to the schedule of hourly and daily operation as follows:

$$Q_{i,j+1}^{Storage} = Q_{i,j}^{Storage} + Q_{i,j+1}^{S} + \alpha Q_{i,j+1}^{H} + \beta Q_{i,j+1}^{B} + \gamma Q_{i,j+1}^{G} - Q_{i,j+1}^{out} - Q_{i,j+1}^{loss} \qquad (5-6)$$

Where indices i, j represent day and hour respectively, the parameter of α, β, γ are equal '1' when a heat source is on and '0' when a heat is off. The priority given to each heat source operation remains unchanged to that given in Table 4.1.

In this optimisation, it was assumed that initially the thermal store carries over an amount of heat from the previous period. This is expressed by the following constraint:

$$Q_{i,j+1}^{Storage} \gg Q^{Intial}$$

$$Q^{Intial} = Q_{i,j}^{Storage} \quad \text{For i=1, j=0}$$

The flow chart algorithm for thermal store capacity is presented in Figure 5.1, with day 1 is taken as January 1st, the range of day 'i' from 1 to 365 are from January 1st to December 31st. For the hour of day 'j' ranging from 0:00 to 07:00, the decision for 'j <=7' means whether the hour is from 0:00 to 7:00. The variation of thermal store capacity during a year can be obtained. The thermal store capacity can be determined by the maximum thermal store capacity as follows:

$$Q_{storage} = \max (Q_{i,j+1}^{Storage}),\ \forall \text{ i, j}$$

Using the previous mathematical formulation, the optimisation model of the thermal store was developed in MATLAB software with data exported to excel to analyses including reliability analysis.

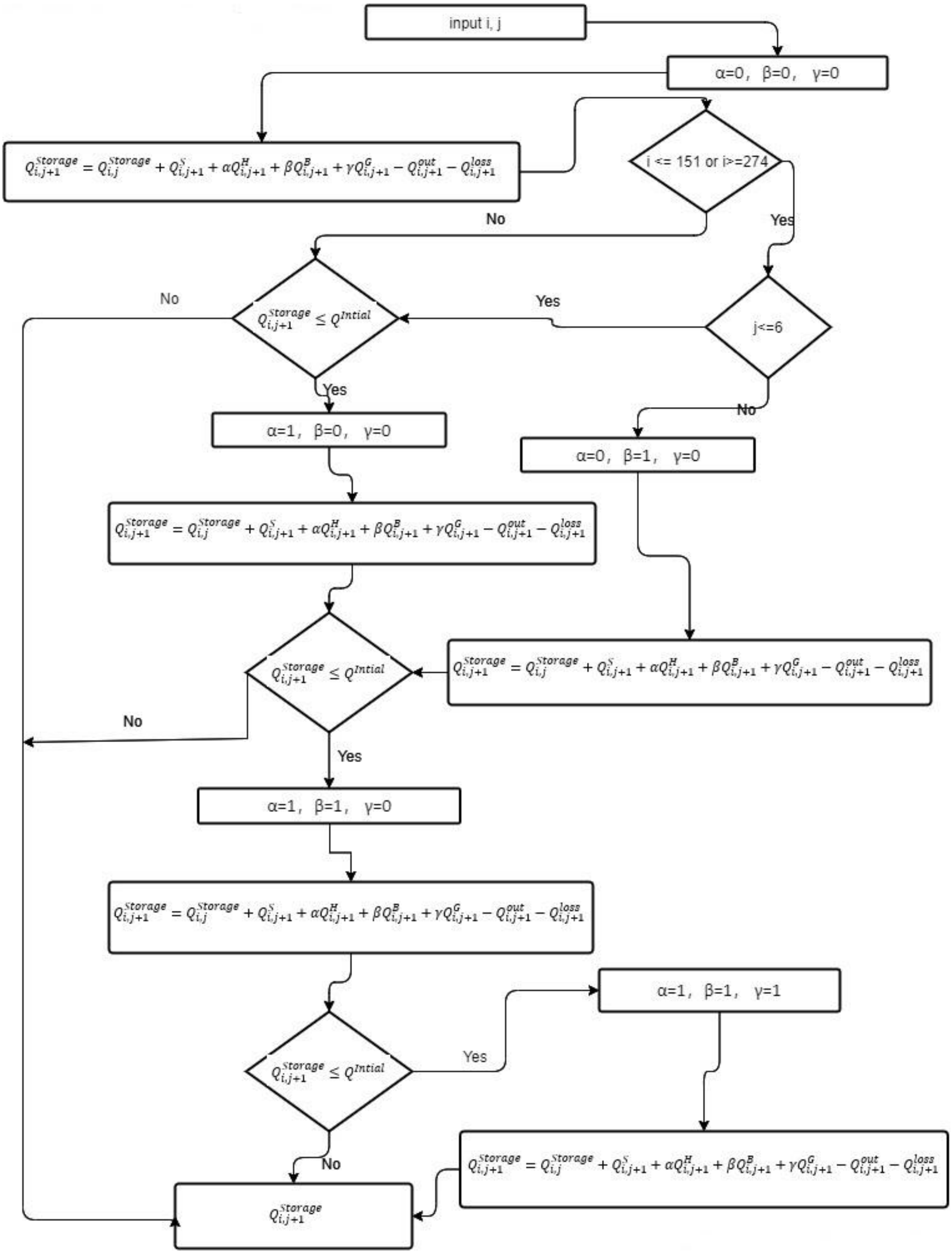


Figure 5.1 A flow chart algorithm for thermal store capacity

5.3 Optimisation of thermal storage capacity

The optimisation of the thermal store heat capacity is important in enhancing energy efficiency, flexibility and accommodation of renewable energy generation into network. To integrate intermittent source of heat from solar collector, a dynamic model based on yearly heat load was used in optimisation of the size of the thermal store.

As described in the previous chapter, four heat sources were selected to supply heat into the thermal store namely gas boiler, biomass boiler, heat pump, and solar thermal collector. The contribution of each heat source is prioritized with solar collector having precedence on other sources. The size of the store is determined as the hourly peak storage capacity over a period of one year required to satisfy the heat demand of the site.

As described in the previous chapter, three sizes solar collector were retained for the optimisation of the thermal store exercise: 1.56 kW, 3.11 kW and 4.67 kW. The thermal store capacity calculation was obtained by considering the dynamic variation of the heat generation and heat load. The MATLAB optimisation codes are given in Appendix C.

5.3.1 Scenario 1: 1.56 kW solar collector

The optimisation process is similar to that described in Chapter 4. According to Chapter 4, the annual thermal capacity of the thermal store with 1.56 kW solar collector where it can be seen that the maximum storage capacity for this scenario is 26.68 kWh. The specific heat capacity of water in thermal store is assumed at 4.18 $kJ/(kg \cdot K)$ and the density of water is 1000 kg/m^3. The volume of thermal store can be estimated according to Equation (5-1), which is 919.12 liters. Based on the value of thermal store volume, the matched manufacturing size can be designed. From the manufacturing data, the tank is cylindrical shapes with 1.03 m diameter, 2.132 m height, and 0.12 m thickness [170].The thermal conductance is 0.041 $W/(m \cdot K)$, and the heat transfer coefficient is 0.3417 $W/(m^2K)$) [170]. The manufacturing data above are used to calculate the variation of thermal store heat loss throughout a year.

Table 5.1 shows a sample of heat generation capacity of heat sources and corresponding thermal store from January 1st. The initial storage capacity of the thermal store is 5 kWh, which is considered the minimum amount of energy storage. For example, at 07:00, the heat load is 8.96 kW, the heat stored carried over from the previous hour is 9.81 kWh, which cannot cover the heat load of the site. Hence, a 10 kW heat pump is brought online for one hour to supply the required heat for the site and store any excess heat in the thermal store which is increased the thermal store capacity to 10.71 kWh at the end of the hour. At 08:00, the heat load increases to 42 kW, and again the 10.71 kWh thermal store capacity cannot cover it, which requires bringing additional capacity by turning on the heat pump (10 kW), biomass boiler (15 kW) and gas boiler (20 kW). The use of the gas boiler is considered as a last priority in terms of carbon emission and is only used to supply peak load. Any heat generation exceedance is then stored in the thermal store which the heat of the thermal store is increased to 13.57 kWh.

Lastly, it is shown in Table 5.1 that for January 1st operation, the gas boiler only was switched on one hour, that heat pump started up for 4 times, while biomass boiler started up twice. The number of startup times of the heat sources is important in that it influences the maintenances cost of the system and decreases reliability. On an annual basis the biomass boiler has the highest startups (935 times), the heat pump (808 times) and gas boiler started up 90 times. This will be explored further in the next chapter.

Table 5.1 Scenario 1 sample thermal storage calculation on January 1st

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	'Demand Met'
01-Jan	00:00								5	1
	01:00	5.79	1.4	0	10	0	0	0.13	9.07	1
	02:00	6.3	0.5	0	10	0	0	0.14	12.64	1
	03:00	7.13	-0.2	0	0	0	0	0.14	5.38	1
	04:00	7.86	-0.7	0	10	0	0	0.14	7.38	1
	05:00	8.43	-0.9	0	10	0	0	0.14	8.81	1
	06:00	8.86	-1.1	0	10	0	0	0.14	9.81	1
	07:00	8.96	-0.4	0	10	0	0	0.14	10.71	1
	08:00	42	0	0	10	15	20	0.14	13.57	1
	09:00	23.2	0.7	0	0	15	0	0.14	5.28	1
	10:00	16.4	1.2	0	10	15	0	0.13	13.75	1
	11:00	14.8	1.7	0	0	15	0	0.13	13.86	1
	12:00	13.9	2.5	0	0	15	0	0.13	14.83	1
	13:00	17.5	2	0	0	15	0	0.13	12.19	1
	14:00	12.8	1.5	0	0	15	0	0.13	14.30	1
	15:00	11	2	0	0	15	0	0.13	18.12	1
	16:00	12.4	1.6	0	0	0	0	0.13	5.59	1
	17:00	14.2	0.8	0	0	15	0	0.14	6.30	1
	18:00	22.4	0.3	0	10	15	0	0.14	8.79	1
	19:00	21	-0.1	0	10	15	0	0.14	12.69	1
	20:00	13.3	-0.2	0	0	15	0	0.14	14.28	1
	21:00	17.4	-0.5	0	0	15	0	0.14	11.71	1
	22:00	17.6	-0.7	0	0	15	0	0.14	8.97	1
	23:00	15.7	-0.8	0	0	15	0	0.14	8.09	1
	24:00	4.74	-0.9	0	0	15	0	0.14	18.21	1

Based on the data from the maximum thermal store capacity that satisfies the heat demand, it can be seen from Table 5.2 that the maximum heat capacity of the thermal store occurs on month July and day 14 which amounts to 25.96 kWh corresponding a thermal store 894.32 liters.

Table 5.2 The thermal store capacity on a representative day during July 14th

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	'Demand Met'
14-Jul	00:00								8.94	1
	01:00	0.00	13.6	0.00	0	0	0	0.10	8.84	1
	02:00	0.00	13.8	0.00	0	0	0	0.10	8.74	1
	03:00	0.00	14.3	0.00	0	0	0	0.10	8.64	1
	04:00	0.00	14	0.00	0	0	0	0.10	8.54	1
	05:00	0.00	14.1	0.00	0	0	0	0.10	8.44	1
	06:00	0.06	15	0.00	0	0	0	0.10	8.28	1
	07:00	13.70	15.7	0.00	10	15	0	0.10	19.48	1
	08:00	2.94	16.4	0.41	0	0	0	0.09	16.85	1
	09:00	0.00	17.4	0.72	0	0	0	0.09	17.47	1
	10:00	0.00	18.1	1.07	0	0	0	0.09	18.45	1
	11:00	0.00	19.4	1.33	0	0	0	0.09	19.70	1
	12:00	0.00	20.3	1.48	0	0	0	0.08	21.09	1
	13:00	0.00	21	1.51	0	0	0	0.08	22.52	1
	14:00	0.00	21.8	1.44	0	0	0	0.08	23.88	1
	15:00	0.00	21.9	1.25	0	0	0	0.08	25.05	1
	16:00	0.00	22.7	0.99	0	0	0	0.08	25.96	1
	17:00	5.89	22	0.63	0	0	0	0.08	20.62	1
	18:00	5.89	20.9	0.11	0	0	0	0.08	14.76	1
	19:00	0.00	19.4	0.00	0	0	0	0.09	14.67	1
	20:00	2.43	18.3	0.00	0	0	0	0.09	12.15	1
	21:00	2.43	16.9	0.00	0	0	0	0.09	9.63	1
	22:00	0.00	15.3	0.00	0	0	0	0.10	9.53	1
	23:00	0.00	14.9	0.00	0	0	0	0.10	9.43	1
	24:00	0.00	14.5	0.00	0	0	0	0.10	9.33	1

Furthermore, the heat supply contribution in terms of the number of operation hours of each source for a duration of one year that can satisfy the demand was evaluated. This is taking into account the UK different electricity tariffs. The night tariffs are those between 00:00 and 07:00, while day tariffs are applied for the remaining time (07:00- 24:00) according to SSE Airtricity [171]. The results of the modelling of the length of operation of each heat source for day and night schedules is shown in Figure 5.2. As it can be observed, the solar collector recorded the highest number of hours of operation with a total of 1891 hours, as it is ranked top for preferential operation. The biomass boiler was the second longest heat source in terms of the number of hours of operation and recording 1775 hours. It was also interesting to notice that the majority of running hours of the biomass boiler were during daytime (86.2%). The heat pump on the other hand contributed more operation hours during night time as electricity were more favorable (780 hours). Finally, given that the gas boiler is used to meet peak load demand, it was operated for only a total of 90 hours with that 90% of time was during day time as this coincides with early morning and late afternoon peak demand for heat.

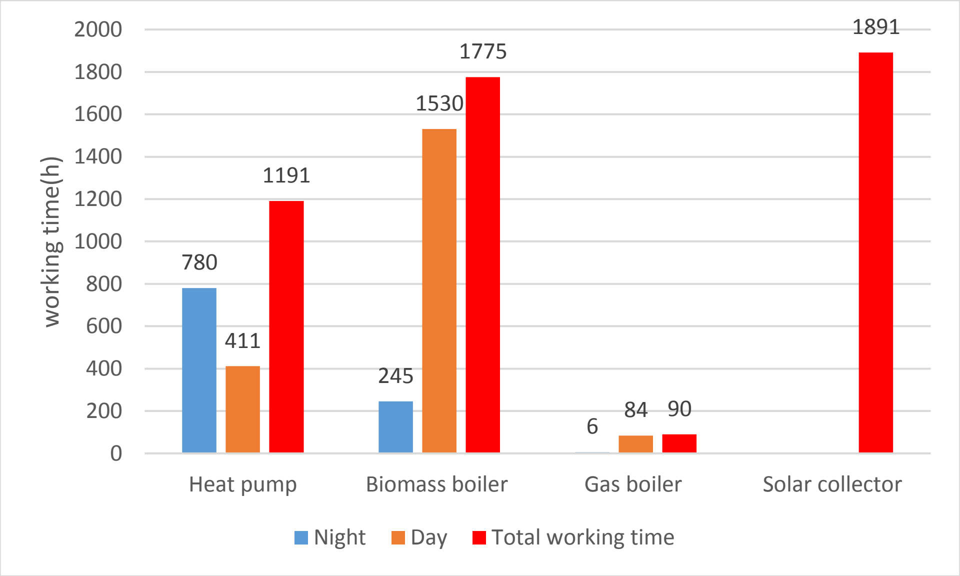


Figure 5.2 Total working time for each heat source in scenario 1.

The site's heat demand is satisfied when there is excess heat storage (i.e., storage capacity higher than 5 kWh) in the thermal store and this indicated by a '1' for 'Demand met' or a '0' otherwise. The 'Demand met' value is evaluated on hourly basis for a full year of operation (8760 hours). The number of hours of operation of which the 'demand met' is '1' is presented as a percentage for which the system would supply heat reliably, as expressed by Equation (4-7). It was found that this contribution of heat sources and thermal store capacity can fulfil the site's heat demand throughout the year, giving a system's reliability of 100%.

A further break down of each heat source contribution in terms of heat generation is shown in Figure 5.3. The largest amount of heat contribution is featured by the biomass boiler, generating a total of 26.625 MWh of heat. The heat pump is the second largest contributor of heat with an annual output of 11.91 MWh, while the contribution of solar collector and gas boiler, though important, is only marginal. The heat pump and biomass boiler are the main heat sources to cover the heat demand, which contribute 93.43% of heat demand.

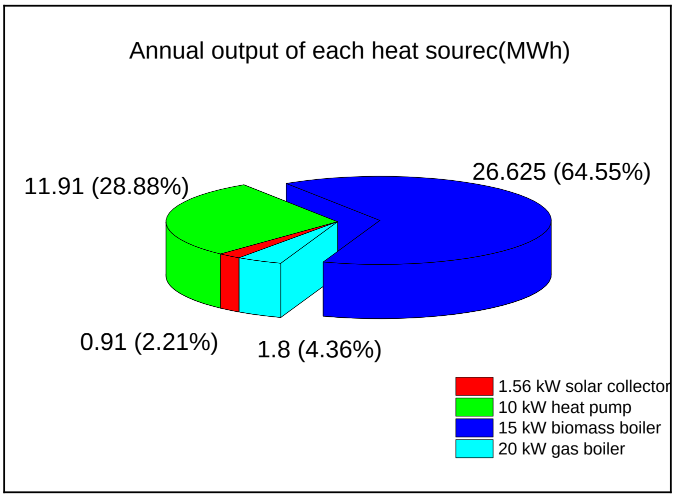


Figure 5.3 Annual output of each heat source in scenario 1

5.3.2 Scenario 2: 3.11 kW solar collector

The similar analytical procedure was repeated for scenario 2. According to Chapter 4, the annual thermal capacity of the thermal store with a 3.11 kW solar collector where it can be seen that the maximum storage capacity for this scenario is 33.12 kWh. Similarly, the volume of thermal store is 1140.98 liters, and the matched manufacturing size can be designed. From the manufacturing data, the tank is of cylindrical shapes with the diameter, height and thickness of 1.24 m, 2.142 m, and 0.12 m, respectively. The thermal conductance is 0.041 $W/(m \cdot K)$, and the overall heat transfer coefficient is 0.3417 $W/(m^2K)$) [170]. The manufacturing data above are used to calculate the variation of thermal store heat loss throughout a year .

Table 5.3 reveals a sample of heat generation capacity of heat sources and corresponding thermal store from January 1st. The initial storage capacity of the thermal store is 5 kWh, which is considered as the minimum amount of energy storage. For example, the heat load is 8.96 kW at 07:00, and the heat stored carried over from the previous hour is 9.61 kWh, which cannot cover the heat load of the site. Hence, a 10 kW heat pump switches for one hour to supply the required heat for the site and store any excess heat in the thermal store which is increased the thermal store capacity to 10.5 kWh at the end of the hour. At 08:00, the heat load increases to 42 kW, and again the 10.5 kWh thermal store capacity cannot cover it, which requires bringing additional capacity by turning on the heat pump (10 kW), biomass boiler (10 kW) and gas boiler (20 kW). The use of the gas boiler is considered as a last priority in terms of carbon emission and is only used to supply peak load. Any heat generation exceedance is then stored in the thermal store which the heat of the thermal store is decreased to 8.33 kWh.

Lastly, it is shown in Table 5.3 that for January 1st operation, the gas boiler only was switched on two hours, that the heat pump started up for 5 times, while biomass boiler started up twice. The number of startup times of the heat sources is important in that it influences the maintenances cost of the system and decreases reliability. On an annual basis the heat pump has the highest startup times (965 times), while the biomass boiler and gas boiler started up for 784 times and 175 times, respectively. This will be explored further in the next chapter.

Table 5.3 Scenario 2 sample thermal storage calculation on January 1st

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	'Demand Met'
01-Jan	00:00								5	1
	01:00	5.79	1.4	0	10	0	0	0.17	9.04	1
	02:00	6.3	0.5	0	10	0	0	0.17	12.6	1
	03:00	7.13	-0.2	0	0	0	0	0.17	5.28	1
	04:00	7.86	-0.7	0	10	0	0	0.17	7.25	1
	05:00	8.43	-0.9	0	10	0	0	0.17	8.65	1
	06:00	8.86	-1.1	0	10	0	0	0.17	9.61	1
	07:00	8.96	-0.4	0	10	0	0	0.17	10.5	1
	08:00	42	0	0	10	10	20	0.17	8.33	1
	09:00	23.2	0.7	0	10	10	20	0.17	25	1
	10:00	16.4	1.2	0	0	0	0	0.17	8.43	1
	11:00	14.8	1.7	0	10	10	0	0.16	13.5	1
	12:00	13.9	2.5	0	0	10	0	0.16	9.44	1
	13:00	17.5	2	0	10	10	0	0.16	11.8	1
	14:00	12.8	1.5	0	0	10	0	0.17	8.85	1
	15:00	11	2	0	0	10	0	0.16	7.64	1
	16:00	12.4	1.6	0	0	10	0	0.17	5.08	1
	17:00	14.2	0.8	0	10	10	0	0.17	10.8	1
	18:00	22.4	0.3	0	10	10	0	0.17	8.21	1
	19:00	21	-0.1	0	10	10	0	0.17	7.08	1
	20:00	13.3	-0.2	0	10	10	0	0.17	13.6	1
	21:00	17.4	-0.5	0	0	10	0	0.17	6.04	1
	22:00	17.6	-0.7	0	10	10	0	0.17	8.26	1
	23:00	15.7	-0.8	0	10	10	0	0.17	12.3	1
	24:00	4.74	-0.9	0	0	0	0	0.17	7.43	1

Based on the data from the maximum thermal capacity of thermal store that satisfies the heat demand, it can be seen from Table 5.4 that the maximum heat capacity of the thermal store occurs on month June and day 6 which amounts to 31.72 kWh corresponding a thermal store 1092.75 liters.

Table 5.4 The thermal store capacity on a representative day during June 6th

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat Loss (kWh)	Storage capacity (kWh)	'Demand Met'
	00:00								5.99	1
	01:00	0.00	7.5	0.00	0	0	0	0.15	5.84	1
	02:00	0.02	7.4	0.00	0	0	0	0.15	5.67	1
	03:00	0.04	7.4	0.00	0	0	0	0.15	5.48	1
	04:00	0.08	7.4	0.00	0	0	0	0.15	5.25	1
	05:00	0.18	7.4	0.00	10	0	0	0.15	14.93	1
	06:00	2.22	7.8	0.00	0	0	0	0.15	12.56	1
	07:00	15.80	9	0.00	10	0	0	0.14	6.62	1
	08:00	3.35	10.5	0.68	10	0	0	0.14	13.81	1
	09:00	0.00	11.8	1.52	0	0	0	0.13	15.20	1
	10:00	0.00	13.3	2.15	0	0	0	0.13	17.23	1
	11:00	0.00	14.2	2.61	0	0	0	0.12	19.71	1
06-Jun	12:00	0.00	16.1	2.88	0	0	0	0.12	22.48	1
	13:00	0.00	16.1	2.90	0	0	0	0.12	25.26	1
	14:00	0.00	16.6	2.72	0	0	0	0.12	27.86	1
	15:00	0.00	17	3.11	0	0	0	0.12	30.08	1
	16:00	0.00	17	1.76	0	0	0	0.12	31.72	1
	17:00	6.31	17	1.03	0	0	0	0.12	26.33	1
	18:00	6.31	16.3	0.30	0	0	0	0.12	20.19	1
	19:00	0.00	14.7	0.00	0	0	0	0.12	20.07	1
	20:00	2.64	12.9	0.00	0	0	0	0.13	17.30	1
	21:00	2.64	11.3	0.00	0	0	0	0.13	14.52	1
	22:00	0.00	10.9	0.00	0	0	0	0.14	14.39	1
	23:00	0.00	10.2	0.00	0	0	0	0.14	14.25	1
	24:00	0.00	9.8	0.00	0	0	0	0.14	14.11	1

Similarly, the heat supply contribution in terms of the number of operation hours of each source for a duration of one year that can satisfy the demand was evaluated. The modelling results of the length of operation of each heat source for day and night schedules are shown in Figure 5.4. As it can be observed, the solar collector became to the second longest heat source in terms of the number of operation hours and recorded same 1891 hours. The biomass boiler recorded the highest number of operation hours with a total of 2107 hours. It is also interesting to notice that the majority of running hours of the biomass boiler are during daytime (88%). The heat pump on the other hand contributed more operation hours during night time as electricity are more favorable (838 hours). Finally, given that the gas boiler is used to meet peak load demand, it was operated for a higher total of 180 hours with that 74.7% of time was during day time.

The similar reliability analytical procedure was repeated, and it was found that this contribution of heat sources and thermal store capacity can fulfil the site's heat demand throughout the year, giving a system's reliability of 100%.

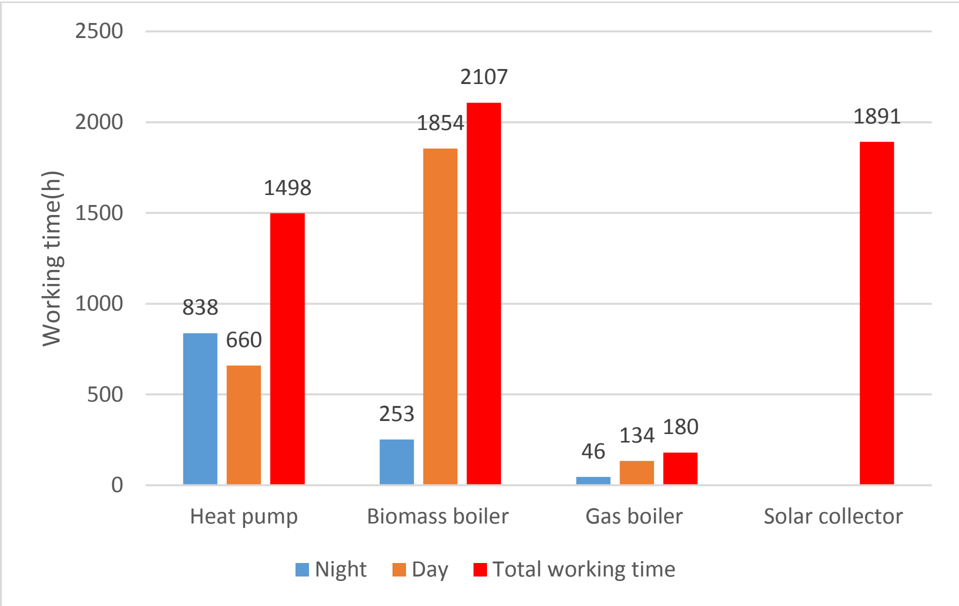


Figure 5.4 Total working time for each heat source in scenario 2.

A further break down of each heat source contribution in terms of heat generation is shown in Figure 5.5. The largest amount of heat contribution is featured by the biomass boiler, generating a total of 21.07 MWh of heat. The heat pump is the second largest contributor of heat with an annual output of 14.98 MWh, while the contribution of the solar collector and gas boiler, though important, is only marginal. The heat pump and biomass boiler are the main heat sources to cover the heat demand, contributing 86.93% heat demand.

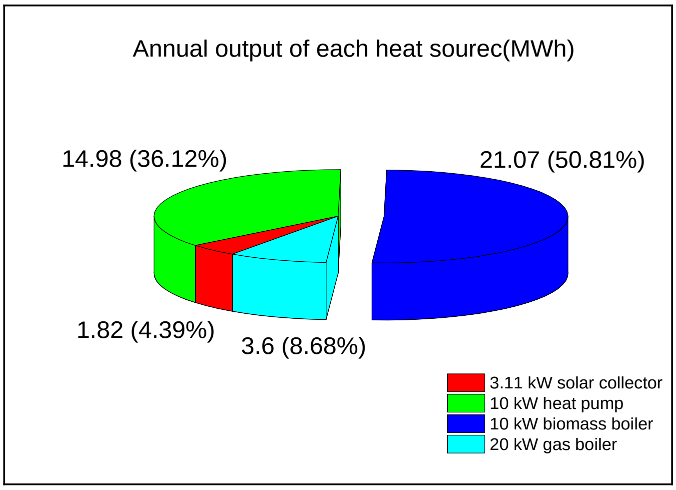


Figure 5.5 Annual output of each heat source in scenario 2

5.3.3 Scenario 3: 4.67 kW solar collector

The similar analytical procedure was repeated for scenario 3. According to Chapter 4, with regard to the annual thermal capacity of the thermal store with a 4.67 kW solar collector, the maximum storage capacity for this scenario is 42.54 kWh. Similarly, the volume of thermal store is 1465.49 liters, and the matched manufacturing size can be designed [170]. The manufacturing data has been described above, which are used to calculate the variation of thermal store heat loss throughout a year.

Table 5.5 reveals a sample of heat generation capacity of heat sources and corresponding thermal store from month February 25th. The initial storage capacity of the thermal store is 5 kWh, which is considered as the minimum amount of energy storage. For example, the heat load is 10.8 kW at 07:00, and the heat stored in previous hour is 6.32 kWh, which cannot cover the heat load of the site. Hence, a 10 kW heat pump switches for one hour to supply the required heat for the site and store any excess heat which decreases the thermal store capacity to 5.35 kWh at the end of the hour. At 08:00, the heat load increases to 44 kW, again the 5.35 kWh thermal store capacity cannot cover it, which requires additional capacity by turning on the heat pump (10 kW), biomass boiler (15 kW) and gas boiler (15 kW). However, the capacity of the thermal storage decreased to 1.16 kWh and the 'Demand met' was '0', which meant that the heat capacity from all the heat generation cannot meet the heat load.

A similar procedure of reliability analysis was repeated, and it was found that this contribution of heat sources and thermal store capacity cannot fulfil the site's heat demand throughout the year, giving a system's reliability of 99.98%.

Table 5.5 Thermal storage calculation with 4.67 kW solar collector, 10 kW heat pump 15 kW biomass boiler and 15 kW gas boiler

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	Demand Met'
25-Feb	00:00	6.36	-2.7	0				0.18	12.9	1
	01:00	7.54	-3.4	0	0	0	0	0.18	5.15	1
	02:00	8.45	-3.9	0	10	0	0	0.18	6.52	1
	03:00	8.97	-3.4	0	10	0	0	0.18	7.37	1
	04:00	9.57	-4.2	0	10	0	0	0.18	7.61	1
	05:00	10.2	-4.6	0	10	0	0	0.19	7.21	1
	06:00	10.7	-4.9	0	10	0	0	0.19	6.32	1
	07:00	10.8	-4.1	0	10	0	0	0.18	5.35	1
	08:00	44	-2.8	0	10	15	15	0.18	1.16	0
	09:00	22	-1.9	0	10	15	15	0.18	19	1
	10:00	15	-0.5	0	0	15	0	0.17	18.8	1
	11:00	10.6	1.9	0	0	0	0	0.16	8.01	1
	12:00	8.76	2	0	0	15	0	0.16	14.1	1
	13:00	10.5	3	0.403	0	15	0	0.16	18.8	1
	14:00	5.72	3.4	0.248	0	0	0	0.16	13.2	1
	15:00	5.72	2.9	0	0	0	0	0.16	7.32	1
	16:00	5.6	3.6	0	0	15	0	0.16	16.6	1
	17:00	7.04	2.5	0	0	0	0	0.16	9.36	1
	18:00	16.9	1.4	0	0	15	0	0.17	7.31	1
	19:00	17	1.1	0	0	15	0	0.17	5.1	1
	20:00	9.57	1.2	0	0	15	0	0.17	10.4	1
	21:00	14	1.4	0	0	15	0	0.17	11.2	1
	22:00	14.2	1.2	0	0	15	0	0.17	11.9	1
	23:00	11.9	1	0	0	15	0	0.17	14.8	1
	24:00	2.73	0.8	0	0	0	0	0.17	11.9	1

Based on the above reliability analysis, the 4.67 kW solar collector, 10 kW heat pump, 15 kW biomass boiler and 15 kW gas boiler cannot satisfy the site's heat demand throughout the year when the heat loss of thermal store is considered into the model. Then the optimisation for different heat generations was again analysed with the thermal store heat loss considered into system. The solar collector is assumed of 4.67 kW is integrated with a thermal store capacity of 2000 L. From the manufacturing data, the tank is of cylindrical shapes with the diameter, height and thickness of 1.44 m, 2.142 m, and 0.12 m respectively. The thermal conductance is 0.041 $W/(m \cdot K)$, and the heat transfer coefficient is 0.3417 $W/(m^2K)$) [170].

Table 5.6 shows storage capacity of 4.67 kW solar collector with different heat generations. The minimum storage capacity of a 4.67 kW solar collector with different generations was 43.34 kWh and the heat pump rated at 10 kW, while the biomass boiler and gas boiler were rated at 10 kW and 25 kW respectively.

Table 5.6 Storage capacity with different heat generation with 4.67 solar collector

Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Storage capacity (kWh)
5	10	30	49.43
5	15	25	46.95
5	20	20	46.95
5	25	10	56.26
10	5	30	49.34
10	10	25	43.34
10	15	20	46.22
10	20	20	51.22
15	5	25	46.47
15	10	20	46.47
15	15	15	46.47
15	20	10	46.47
15	25	5	50.15
20	5	20	53.34
20	10	20	51.22
20	15	5	51.97
25	5	15	51.79
25	10	5	56.27

Table 5.7 shows a sample of heat generation capacity of heat sources and corresponding thermal store from January 1st. The initial storage capacity of the thermal store is 5 kWh, which is considered as the minimum amount of energy storage. For example, the heat load is 8.96 kW at 07:00, and the heat stored in the previous hour is 9.42 kWh, which cannot cover the heat load of the site. Hence, a 10 kW heat pump switches for one hour to supply the required heat for the site and store any excess heat which increases the thermal store capacity to 10.26 kWh at the end of the hour. At 08:00, the heat load increases to 42 kW, and again the 10.26 kWh thermal store capacity cannot cover it, which requires bringing additional capacity by turning on the heat pump (10 kW), biomass boiler (10 kW) and gas boiler (25 kW). The use of the gas boiler is considered as a last priority in terms of carbon emission and is only used to supply peak load. Any heat generation exceedance is then stored in the thermal store which increases to 13.07 kWh.

Lastly, it is shown in Table 5.7 that on January 1st operation, the gas boiler only was switched on two hours, and that the heat pump started up for 7 times, while the biomass boiler started up twice. The number of startup times of the heat sources is important in that it influences the maintenances cost of the system and decreases reliability. On an annual basis the heat pump has the highest startup times (946 times), while the biomass boiler and gas boiler started up for 875 times and 186 times, respectively. This will be explored further in the next chapter.

Table 5.7 Scenario 3 sample thermal storage calculation on January 1st

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	'Demand Met'
01-Jan	00:00								5.00	1
	01:00	5.79	1.4	0	10	0	0	0.20	9.01	1
	02:00	6.3	0.5	0	10	0	0	0.20	12.52	1
	03:00	7.13	-0.2	0	0	0	0	0.20	5.19	1
	04:00	7.86	-0.7	0	10	0	0	0.20	7.12	1
	05:00	8.43	-0.9	0	10	0	0	0.21	8.49	1
	06:00	8.86	-1.1	0	10	0	0	0.21	9.42	1
	07:00	8.96	-0.4	0	10	0	0	0.20	10.26	1
	08:00	42	0	0	10	10	25	0.20	13.07	1
	09:00	23.2	0.7	0	10	10	0	0.20	9.71	1
	10:00	16.4	1.2	0	10	10	0	0.20	13.12	1
	11:00	14.8	1.7	0	0	10	0	0.20	8.16	1
	12:00	13.9	2.5	0	10	10	0	0.19	14.07	1
	13:00	17.5	2	0	0	10	0	0.19	6.36	1
	14:00	12.8	1.5	0	10	10	0	0.20	13.42	1
	15:00	11	2	0	0	10	0	0.19	12.17	1
	16:00	12.4	1.6	0	0	10	0	0.20	9.58	1
	17:00	14.2	0.8	0	0	10	0	0.20	5.23	1
	18:00	22.4	0.3	0	10	10	25	0.20	27.66	1
	19:00	21	-0.1	0	0	0	0	0.20	6.49	1
	20:00	13.3	-0.2	0	10	10	0	0.20	13.02	1
	21:00	17.4	-0.5	0	0	10	0	0.20	5.38	1
	22:00	17.6	-0.7	0	10	10	0	0.20	7.58	1
	23:00	15.7	-0.8	0	10	10	0	0.20	11.63	1
	24:00	4.74	-0.9	0	0	0	0	0.21	6.68	1

Based on the data from the maximum thermal capacity of thermal store that satisfies the heat demand, it can be seen from Table 5.8 that the maximum heat capacity of the thermal store occurs on July 5th which amounts to 43.34 kWh corresponding a thermal store 1492.36 liters.

Table 5.8 The thermal store capacity on a representative day during July 5th

Hourly Average		Heat load (kW)	Ambient temperature (℃)	Solar collector (kW)	Heat pump (kW)	Biomass boiler (kW)	Gas boiler (kW)	Heat loss (kWh)	Storage capacity (kWh)	'Demand Met'
5-Jul	00:00								9.92	1
	01:00	0.00	14.4	0.00	0	0	0	0.15	9.77	1
	02:00	0.00	13.7	0.00	0	0	0	0.15	9.62	1
	03:00	0.00	13.6	0.00	0	0	0	0.15	9.47	1
	04:00	0.00	13.4	0.00	0	0	0	0.15	9.32	1
	05:00	0.00	12.6	0.00	0	0	0	0.15	9.17	1
	06:00	0.00	14	0.00	0	0	0	0.15	9.02	1
	07:00	13.89	16.3	0.31	10	0	0	0.14	5.30	1
	08:00	2.98	18.4	1.47	10	0	0	0.13	13.66	1
	09:00	0.00	21	2.57	0	0	0	0.12	16.10	1
	10:00	0.00	22.4	3.42	0	0	0	0.12	19.40	1
	11:00	0.00	23.8	4.11	0	0	0	0.11	23.40	1
	12:00	0.00	24.5	4.58	0	0	0	0.11	27.87	1
	13:00	0.00	25.4	4.67	0	0	0	0.10	32.44	1
	14:00	0.00	25.2	4.42	0	0	0	0.10	36.75	1
	15:00	0.00	24.3	3.76	0	0	0	0.11	40.39	1
	16:00	0.00	24	3.06	0	0	0	0.11	43.34	1
	17:00	5.97	22.9	2.03	0	0	0	0.11	39.29	1
	18:00	5.97	21.9	0.90	0	0	0	0.12	34.10	1
	19:00	0.00	20.8	0.00	0	0	0	0.12	33.98	1
	20:00	2.47	19.6	0.00	0	0	0	0.13	31.38	1
	21:00	2.47	18.6	0.00	0	0	0	0.13	28.79	1
	22:00	0.00	17.7	0.00	0	0	0	0.13	28.65	1
	23:00	0.00	17	0.00	0	0	0	0.14	28.51	1
	24:00	0.00	16.6	0.00	0	0	0	0.14	28.37	1

Similarly, the heat supply contribution in terms of the number of operation hours of each source for one year that can satisfy the demand was evaluated. The modelling results of operation length of each heat source for day and night schedules are shown in Figure 5.6. As it can be observed, the solar collector became to the second longest heat source in terms of the number of operation hours, amounting to 1891 hours. The biomass boiler recorded the highest number of operation hours, with a total of 1998 hours. It is also interesting to notice that the majority of running hours of the biomass boiler are during daytime (87.5%). The heat pump on the other hand contributed more operation hours during night time as electricity are more favorable (830 hours). Finally, given that the gas boiler is used to meet peak load demand, it worked for a higher total of 187 hours, of which 73.8% of time was during day time.

The similar reliability analytical procedure was repeated, and it was found that this contribution of heat sources and thermal store capacity can fulfil the site's heat demand throughout the year, with the system's reliability of 100%.

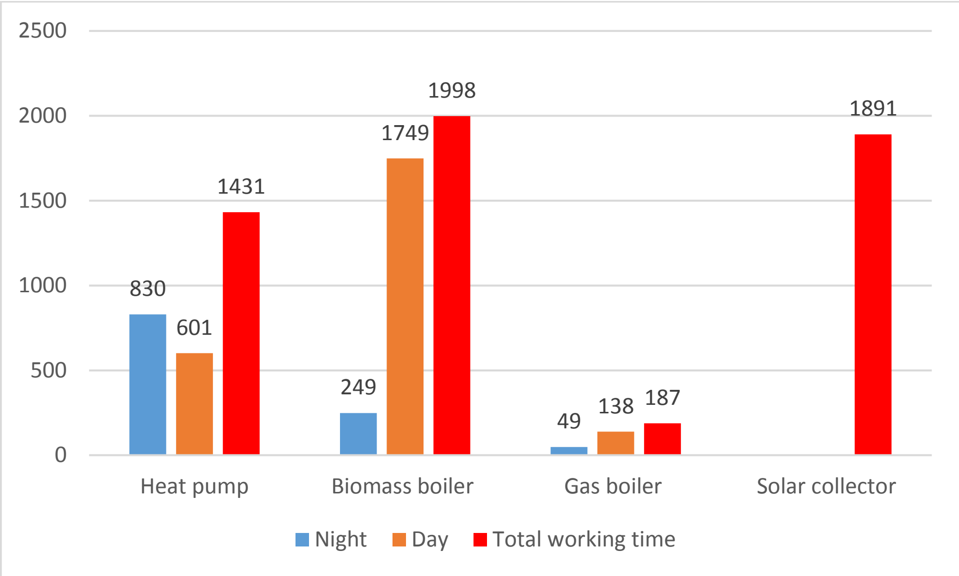


Figure 5.6 Total working time for each heat source in scenario 3

A further break down of each heat source contribution in terms of heat generation is shown in Figure 5.7. The largest amount of heat contribution is featured by the biomass boiler, generating a total of 19.98 MWh of heat. The heat pump is the second largest contributor of heat with annual an output of 14.31 MWh, while the contribution of the solar collector and gas boiler, though important, is only marginal. The heat pump and biomass boiler are the main heat sources to cover the heat demand, contributing 82.22% of heat demand.

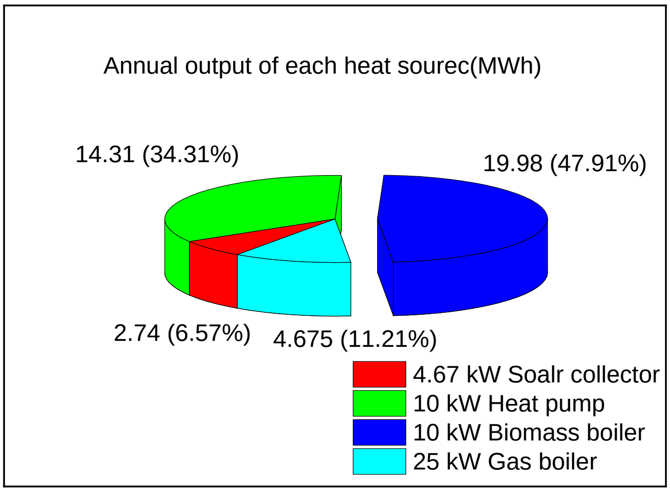


Figure 5.7 Annual output of each heat source in scenario 3

5.4 The impact of heat loss in thermal storage for LTDH

As described in the previous section, three sizes of solar collectors were retained for the optimisation of the thermal store exercise: 1.56 kW, 3.11 kW and 4.67 kW. The heat loss of thermal store with three sizes of solar collectors is shown in Figure 5.8. As the heat capacity of solar collector increased, the heat loss of thermal store increased. The heat loss of thermal store in scenario 1 was the lowest with the total heat loss of 986.19 kWh, while the heat loss of thermal in scenario 3 increased by 46.64% with highest heat loss of 1446.19 kWh. As the solar collector output increased, the thermal store capacity increased. As a result, both the volume of thermal store and heat loss increased.

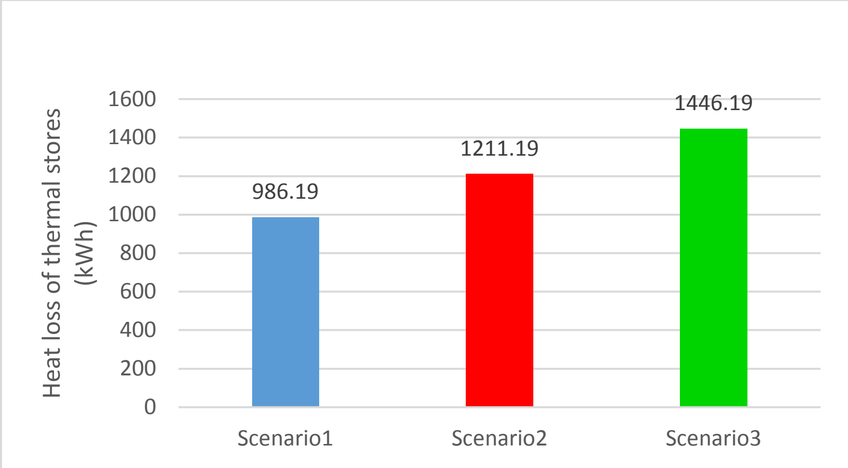
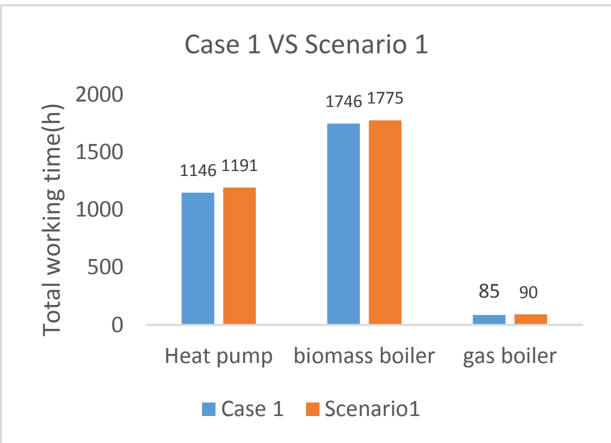


Figure 5.8 The heat loss of thermal store in three scenarios.

As the storage heat loss model has been established, the operation results for LTDH system can be obtained. The operation results for LTDH system without heat loss of thermal storage have been obtained in Chapter 4. The results can be compared to analyse the storage heat loss impact on the LTDH system.

There are three cases in Chapter 4, while there are three scenarios in Chapter 5. However, case 3 and scenario 3 are different due to the different outputs of the biomass boiler and gas boiler. The comparison can be made for case 1 VS scenario 1 and case 2 VS scenario 2., because the only difference in the cases and scenarios is heat loss.



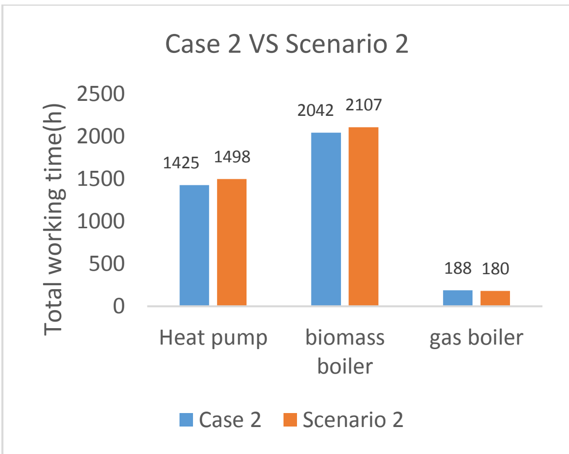


Figure 5.9 The total working time comparison for Case 1 VS Scenario 1, Case 2 VS Scenario 2

The comparison for total working time can be seen in Figure 5.9. The total working time for biomass boiler and heat pump in scenario 1 increased by 1.66% and 3.92% respectively, compared with case 1. Meanwhile, the total working time for the biomass boiler increased by 3.18% from case 2 to scenario 2, while the heat pump increased by 5.12%. As discussed above, the heat pump and biomass boiler are the main heat generations for the LTDH system. When the heat loss of thermal storage is considered in the LTDH system, the main heat generation need to work for more time.

The storage capacity for case 1, scenario 1, case 2 and scenario 2 can be seen in Figure 5.10. The thermal store capacity in case 1 decreased from 26.68 kWh to 25.96 kWh in scenario 1 and the thermal store capacity decreased by 4.23% from case 2 to scenario 2. Generally, the choice of thermal storage should take into account safety factors, the volume of storage will be a bit larger. Overall, the storage capacity would decrease when the thermal storage is considered into the LTDH system. This can help to prevent the selection of oversized thermal store.

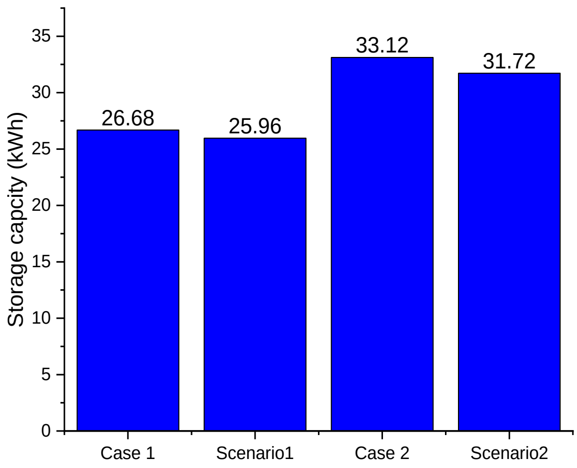


Figure 5.10 The storage capacity comparison for Case 1 VS Scenario 1, Case 2 VS Scenario 2

5.5 Summary

The developed methodology was considered for the thermal storage heat loss, the new model for the LTDH with storage heat loss has been established. According to the previous three cases, the storage volume can be determined. Hence, the storage capacity for three scenarios have been optimised according to different outputs of solar collectors after the reliability analysis. The optimisation for heat sources and thermal store has been obtained, which can be divide into three scenarios.

The maximum thermal store capacity for scenario 1 is 25.96 kWh, while the heat generations is 1.56 kW solar collector, 10 kW heat pump, 15 kW biomass boiler and 20 kW gas boiler in LTDH system.

The maximum thermal store capacity for scenario 2 is 31.72 kWh, while the heat generations is for 3.11 kW solar collector, 10 kW heat pump, 10 kW biomass boiler and 20 kW gas boiler in LTDH system.

The maximum thermal store capacity for scenario 3 was 43.34 kWh, while the heat generations is 4.67 kW solar collector, 10 kW heat pump, 10 kW biomass boiler and 25 kW gas boiler in LTDH system.

The total working time for each heat source has been obtained in three scenarios including day time and night time. The total working time of solar collector for three scenarios is same at 1891 hours. Excluding the solar collector working time, the biomass boiler is the longest operation device in three scenarios, then heat pump is the second longest operation device. The last one is gas boiler. The biomass boiler is the largest heat generation in three scenarios, while the second largest is heat pump. The last two is solar collector and gas boiler. The biomass boiler and heat pump contribute more than 80% of total heat demand, as a result they are the main heat generations. The start-up time of biomass boiler, heat pump and gas boiler in three scenarios have been obtained as well as total working time and annual output. This will be explored in the next chapter for environmental and economic analysis.

As the heat capacity of solar collector increased, the heat loss of thermal store increased. The thermal store heat loss with the 4.67 kW solar collector increased by 46.64% compared with the 1.56 kW solar collector. As the thermal store heat loss is considered into dynamic analysis, the working time of main heat generations increased. However, the maximum thermal store capacity during a year decreased, which can help to prevent the selection of oversized thermal store.

6 Economic and environmental analysis

6.1 Introduction

This chapter analyses the economic and environmental feasibility of the LTDH investigated in this project. The cost of operation and capital of each heat generation technology is evaluated under the three design scenarios. The environment impact of each technology and system in regard to carbon emission.

6.2 Methodology

A mathematical formulation of dispatching heat to the low temperature heat network at optimal operation cost was developed in line with the current energy policy, the UK government implements Renewable Heat Incentive (RHI) scheme to encourage adoption of renewable energy. In this work, the economic analysis takes into account the benefit of government subsidies for renewable heat source. Furthermore, the government introduces carbon emission taxation to simulate energy efficiency practices in industry and keeps with the zero emission targets.

The feasibility analysis of district heating system considers many parameters which include both operation, maintenance and environmental impact. Amongst these are heat generation source operation and running time schedule, operational priority, operation efficiency, heat generation output capacity, start-up period, running off period, carbon emission taxation and maintenance [172]. The annual operation cost for each heat generation is determined taking into account the following: total cost of fuel/power consumption, annual maintenance cost, and government subsidies and financial incentives (i.e. RHI).

In this analysis it was assumed that all the heat generating plants are located in proximity to each other (e.g., house in an energy center) and heat is then distributed to end users forming a small community. As described in previous chapter, the main heat generating technologies considered in this work are solar thermal collector, heat pump, biomass and gas boiler.

6.2.1 Solar collector

As the primary source of heat supply to a solar thermal collector is solar incident on the collector, the cost of which is zero. However, the dispatch of solar hot water supply is intermittent in nature as it depends on time and location site. In the current analysis, the hourly heat generation profile of solar collector was quantified and established in Chapter 3. In this model, solar collector takes operational precedence as the energy is free and benefits from government subsidies in the form of RHI. The operation cost of the solar system will be based on the actual heat supplied from solar thermal system which in turns reflects on maintenance cost incurred. As the fuel cost is zero and taking into account the payment under RHI, the annual cost of generation heat from the system can be expressed as:

$$F_S = F_{s-m} - F_{s-gs} \tag{6-1}$$

Where F_S is the annual operation cost for solar collector (£), F_{s-m} is the annual maintenance for solar collector (£) and F_{s-gs} is the annual government subsidies (£).

The government subsidies are evaluated on the basis of the annual amount of heat generated and can be written as:

$$F_{s-gs} = f_{s-gs} Q_{s-out} \tag{6-2}$$

Where Q_{s-out} is the total heat output of the solar collector (kWh), f_{s-gs} is solar collector tariff of government subsidy (£/kWh).

6.2.2 Biomass boiler

The yearly operation cost for biomass boiler includes the total operation cost of its energy fuel consumption, electricity cost and annual maintenance cost, F_{b-m}. In addition, the biomass is considered a renewable fuel and qualifies for government subsidies of heat generated.

The amount of heat generation from the biomass per year can be expressed as

$$Q_{b-out} = \eta_b m_{b1} LHV_b \qquad (6-3)$$

Where $Q_{b-\text{out}}$ is the total heat output of biomass boiler (kWh), η_b is the efficiency of biomass boiler, LHV_b is the lower heating value of fuel (kWh/kg), m_{b1} is the mass of fuel burnt in the boiler (kg). The efficiency of the biomass boiler is assumed to be constant in this analysis even though the load factors and fuel moisture content can influence it greatly, as explained in Chapter 2.

Another consideration of biomass boiler system performance is the energy usage during startup period. Compared to conventional gas boiler, biomass boiler water circuit contains a large quantity of water, thus the thermal inertia and starting period can be longer resulting additional energy waste. The energy required to heat up the internal water circuit during startup period can be calculated as the cost during start-up in which the whole system's temperature is brought to the required level (i.e., 65 °C). This start up energy can be expressed as follows:

$$Q_{b-cw} = m_{b-cw} c_p \frac{\Delta T}{3600} \Delta t_{b2} \qquad (6-4)$$

Where Q_{b-cw} is heat energy for internal water circuit (kWh), m_{b-cw} is the mass of internal circuit water (kg), c_p is specific heat of water [kJ/(kg K)], ΔT is the temperature difference between initial cold water temperature and supply water temperature, Δt_{b2} is the total number of startups. The total fuel consumption during start up periods can also be determined from the estimated amount of fuel used as:

$$Q_{b-cw} = \eta_b m_{b2} LHV_b \qquad (6-5)$$

Where m_{b2} is the mass of fuel for star-up period (kg).

Therefore, the total cost of fuel consumption is expressed:

$$F_{biomass} = f_b(m_{b1} + m_{b2}) \qquad (6-6)$$

Where the $F_{biomass}$ is the cost for fuel (£), f_b is fuel price per kilogram (£/kg).

The automated biomass fuel mechanical feeder is driven by electric motors which in turn consume non-negligible power. In this analysis two electricity tariffs are considered. The cost of running electricity auxiliary equipment can be calculated from:

$$F_{electrcity} = f_{elec-day}E_b\Delta t_{b-d} + f_{elec-night}E_b\Delta t_{b-n} \qquad (6-7)$$

Where $F_{electrcity}$ is the cost of electricity (£), $f_{elec-day}$ is the standard electricity price (£/kWh), $f_{elec-night}$ is the off-peak electricity price (£/kWh), Δt_{b-d} is total working hour during daytime (h), Δt_{b-n} is total working hour during off-peak night (h), E_b is the output of biomass boiler (kW).

Therefore, the yearly biomass boiler operational cost is expressed as follows:

$$F_B = F_{biomass} + F_{electrcity} + F_{b-m} - F_{b-gs} \qquad (6-8)$$

Where F_{b-gs} represents the government subsidies in the form of RHI which paid to the boiler operator. This is written as:

$$F_{b-gs} = f_{b-gs}Q_{b-out} \qquad (6-9)$$

Where f_{b-gs} is the biomass boiler tariff of government subsidy (£/kWh).

From Equation (6-6), (6-7), (6-8) and (6-9) which expresses the annual cost of running the biomass boiler can be written as:

$$F_B = f_b(m_{b1} + m_{b2}) + f_{elec-day}E_b\Delta t_{b-d} + f_{elec-night}E_b\Delta t_{b-n} + F_{b-m} - f_{b-gf}Q_{b-out} \quad (6-10)$$

6.2.3 Gas boiler

The procedure of calculating the yearly operational cost of the gas boiler is similar to that of the biomass boiler presented previously with the exception of RHI which the gas boiler does not qualify for. Therefore, the annual cost includes the total cost of its energy fuel consumption, electricity cost and annual maintenance cost.

The total cost of fuel, F_g, used to generate heat using the gas boiler can be expressed as follows:

$$F_g = F_{gas} + F_{electrcity} + F_{g-m} \qquad (6-11)$$

Where $F_{gas} = f_g * \left(Q_{g-input+}Q_{g-cw}\right)$ is the fuel cost (£) used in normal operating hours output ($Q_{g-input} = \frac{Q_{g-out}}{\eta_g}$) and start up time fuel ($Q_{g-cw} = m_{g-cw}c_p \frac{\Delta T}{3600} \Delta t_{g2}$), $F_{electrcity}$ is the cost of electricity (£) used to drive the water circulating pump and F_{g-m} is the maintenance cost, f_g is the gas price (£/kWh), $Q_{g-input}$ is the input energy of gas (kWh), η_g is the efficiency of gas boiler, Q_{g-cw} is the total energy used at startup periods with m_{g-cw} the mass of internal circuit water (kg), c_p is specific heat of water [kJ/(kg K)], ΔT is the temperature difference between initial cold water temperature and supply water temperature and), Δt_{g2} is the total number of startups.

6.2.4 Heat pump

The annual operation cost for running heat pump includes the total cost of power supply and the annual maintenance cost. As with solar collector and biomass boiler, heat pump is considered as a renewable energy system and hence benefits from government financial subsidy of RHI. As explained in Chapter 2, the heat pump thermal performance is usually specified by the seasonal coefficient of performance (COP), which is a measure of how much heat is generated per unit of electrical power input. The COP is given by:

$$COP = \frac{Q_p}{Q_e} \qquad (6-12)$$

Where Q_p is the heat pump output (kW), Q_e is the electricity input (kW), COP is the coefficient of performance for heat pump.

Therefore, considering the normal and off-peak operating time and associated electricity tariffs for each period, the annual cost of electrical power consumed is given by:

$$F_{electrcity} = f_{elec-day}Q_e\Delta t_{h-d} + f_{elec-night}Q_e\Delta t_{h-n} \qquad (6-13)$$

Where $F_{electrcity}$ is electricity cost (£), Δt_{h-d} is heat pump working hour for day time, Δt_{h-n} is heat pump working hour for off-peak night time.

The government subsidies for the heat generated by the heat pump (F_{h-gs}) can be calculated by the following

$$F_{h-gs} = f_{h-gs}(Q_p - Q_e)(\Delta t_{h-d+}\Delta t_{h-n}) \qquad (6-14)$$

Where f_{h-gs} is the heat pump tariff of government subsidy (£/kWh). The government subsidies only applies to the renewable energy of the heat generated which is $Q_p - Q_e$.

Taking into account the cost of maintenance of the heat pump, the annual cost of running the heat pump becomes:

$$F_H = F_{electrcity} + F_{h-m} - F_{h-gs} \qquad (6-15)$$

The annual cost of running the heat pump given by Equation (6-15) can be expressed using Equation (6-13), (6-14) as:

$$F_H = f_{elec-day}Q_e\Delta t_{h-d} + f_{elec-night}Q_e\Delta t_{h-n} + F_{h-m} - f_{h-gs}(Q_p - Q_e)(\Delta t_{h-d+}\Delta t_{h-n}) \quad (6-16)$$

6.2.5 Water pump

The heat network requires water pump to circulate the heat from the heat sources to the end user. In this analysis it was assumed that each heat source has a dedicated water circulating pump. To calculate the operation cost of water circulating pump, water circulation capacity is selected by its maximum flow rate:

$$m_{cw-max} = \frac{Q_{load-max}3600}{C_p(T_S - T_R)} \qquad (6-17)$$

Where m_{cw-max} is the mass flow rate of water pump (kg/h), $Q_{load-max}$ is the maximum heat load (kW), T_S is the supply water temperature (°C), T_R is the return water temperature (°C).

From the site's heat demand $Q_{d,i}$, the hourly mass flow rate can be expressed as:

$$m_{cw-d,i} = \frac{Q_{d,i}3600}{C_p(T_S - T_R)1000} \qquad (6-18)$$

Where $m_{cw-d,i}$ is the hourly mass flow rate (m^3/h).

The annual cost of running the water pump can be calculated as:

$$F_{W-d,i} = \sum_{d=1}^{365}(\sum_{i=1}^{i=7} f_{elec-night}\, P_{elec-d,i} + \sum_{i=8}^{i=24} f_{elec-\mathrm{day}}\, P_{elec-d,i}) \qquad (6-19)$$

Where $P_{elec-d,i}$ is the hourly power input (kW).

6.2.6 Thermal store

Since the solar hot water generation is available only in sunny hours during day time and which does not coincide with peak heat demand pattern of early morning or late afternoon, thermal store allows to store the deploy this heat when needed. In this analysis, the operation cost of storing heat in a thermal store is calculated as follows:

$$F_{th} = \sum_{t=1}^{8640} q_{th-t}\, f_{th} \qquad (6-20)$$

Where F_{th} is the total thermal storage maintenance cost (£), q_{th-t} is hourly thermal storage capacity (kWh), f_{th} is the maintenance cost (£/kWh).

6.2.7 Total carbon emission and total cost for the LTDH system

The carbon emission mainly from heat pump and gas boiler, meanwhile biomass boiler also can produce a little of carbon emission. Solar energy is completely clean energy, which would not produce carbon emission. The total carbon emission can be expressed

$$M_{total} = M_B + M_G + M_H \qquad (6-32)$$

Where M_B is the annual carbon emission from biomass boiler (kg), M_G is the annual carbon emission from gas boiler (kg), M_H is the annual carbon emission from heat pump (kg).

The carbon emission can be calculated by the consumption of fuel. However, the carbon emission can also be calculated by the total output of each heat source in this study. The carbon emission from each heat source can be described as follows:

$$M_B = Q_{b-out}m_B \qquad (6-33)$$

$$M_G = Q_{g-out}m_G \qquad (6-34)$$

$$M_H = \frac{Q_{h-out}}{COP}m_H \qquad (6-35)$$

Where Q_{b-out} is the total output of biomass boiler (kWh), Q_{g-out} is the total output of gas boiler (kWh), Q_{h-out} is the total output of heat pump (kWh), m_B is carbon emission per kilowatt hour for biomass boiler (kg/kWh), m_G is carbon emission per kilowatt-hour for gas boiler (kg/kWh), m_H is carbon emission per kilowatt-hour for heat pump (kg/kWh).

Equation (6-32) combined with Equation (6-33), (6-34), (6-35) can be expressed:

$$M_{total} = Q_{b-out}m_B + Q_{g-out}m_G + Q_{h-out}m_H \qquad (6-36)$$

As the previous section described, the cost of the LTDH system (F_{total}) with multiple heat sources include solar collector cost, biomass boiler cost, gas boiler cost, heat pump cost, water pump cost, thermal storage cost and carbon emission taxation, which can be expressed by the following equation:

$$F_{total} = \sum_{n} F_n \qquad (6-37)$$

$$n = S, B, G, th, H.W, taxation$$

6.3 Analysis and operating parameters

From the heat generation optimisation of Chapter 5 where three design scenarios were investigated which include the following:

- Scenario 1: 1.56 kW solar collector, 10 kW heat pump, 15 kW biomass boiler, 20 kW gas boiler and a thermal store of 0.894 m^3(or 25.96 kWh)
- Scenario 2: 3.11 kW solar collector, 10 kW heat pump, 10 kW biomass boiler, 20 kW gas boiler and a thermal store of 1.093 m^3(or 31.72 kWh)
- Scenario 3: 4.67 kW solar collector, 10 kW heat pump, 10 kW biomass boiler, 25 kW gas boiler and a thermal store of 1.492 m^3(or 43.34 kWh)

The annual qualifying heat generated by each technology of different design scenarios is recapitulated in Table 6.1.

Table 6.1 Total output for each heat source.

	Solar energy (MWh)	Heat pump (MWh)	Biomass boiler (MWh)	Gas boiler (MWh)
Scenario 1	0.91	11.91	26.625	1.8
Scenario 2	1.82	14.98	21.07	3.6
Scenario 3	2.74	14.31	19.98	4.675

In this case study, the government subsidies of RHI was considered. As the case study represents an aggregation of households forming a community, non-domestic RHI rate was applied. The current non-domestic rates of RHI payments for solar collector, air source heat pump and biomass boiler are summarised in Table 6.2.

Table 6.2 Non-domestic RHI tariff rates that apply for installation with an accreditation date on or after 1 July 2017 [125]

Type of technology	**Heat output capacity**	**Tariff(£/kWh)**
Solar collector	< 200kWth	0.1075
Air source heat pump	All capacities	0.0269
Biomass boiler	< 200kWth Tier 1*	0.0279
	< 200kWth Tier2*	0.0073

* In each year the Tier 1 tariff is paid until the system has operated up to 15% of the annual rated output and the rest of the output in the year, the Tier 2 tariff will apply

The equipment design specification outlined in Chapter 5 for the three case scenarios are then checked with existing manufactures data as shown in Table 6.3. The unit cost (£/kWh) of maintenance of heat pump, biomass boiler and gas boiler are strongly dependent on the amount of heat generated or the number of running hours per year. The maintenance cost of solar collector is however often estimated and obtained from existing literature with a range of 0.9 to 1.8% of capital costs [173]. The maintenance cost was also related to topping up the system with anti-freezing solution every 3-5 years at £140 [174, 175]. The maintenance cost of storing heat in the thermal store is also influenced by hourly thermal store capacity.

Table 6.3 The manufactural data for devices in LTDH

	Heat source technology	Fuel type	Data specification	Reference
Scenario 1:	Biomass	Wood chips	Capacity: 2.9-12.9 kW, Water content: 117L, Efficiency: 92.8%, Electricity: 73 W, Maintenance: £100	[176]
	Gas boiler	Gas	Max output: 24 kW, Efficiency: 92%, Electricity: 17 W, Water content: 3.9L, Maintenance: £275	[177]
	Heat pump	Electricity	Heating capacity:16kW, COP: 3.6, Refrigerant: R470C, Maintenance: £360	[178]
	Solar collector	Solar energy	Maintenance: £65	[175]
	Thermal storage	-	Maintenance: 0.001£/kWh	[179]
Scenario 2:	Biomass	Wood pellets	Capacity: 4.4-14.9 kW, Water content: 27L, Efficiency: 95.7%, Electricity: 66 W, Maintenance: £100	[180]
	Gas boiler	Gas	Max output: 24 kW, Efficiency: 92%, Electricity: 17 W, Water content: 3.9L, Maintenance: £275	[177]
	Heat pump	Electricity	Heating capacity:16kW, COP: 3.6, Refrigerant: R470C, Maintenance: £360	[178]
	Solar collector	Solar energy	Maintenance: £95	[175]
	Thermal storage	-	Maintenance: 0.001£/kWh	[179]
Scenario 3:	Biomass	Wood pellets	Capacity: 4.4-14.9 kW, Water content: 27L, Efficiency: 95.7%, Electricity: 66 W, Maintenance: £100	[180]
	Gas boiler	Gas	Max output: 30 kW, Efficiency: 92%, Electricity: 19W, Water content: 3.9L, Maintenance: £275	[181]
	Heat pump	Electricity	Heating capacity:16kW, COP: 3.6, Refrigerant: R470C, Maintenance: £360	[178]
	Solar collector	Solar energy	Maintenance: £125	[175]
	Thermal storage	-	Maintenance: 0.001£/kWh	[179]

The applied primary fuel type and tariffs used by the different technologies in this analysis is recapitulated in Table 6.4. Two type of biomass fuels are listed- wood chips and wood pellets which cost depend on the production process. Similarly, two type prices of electricity are considered- standard rate (daytime) tariff and off-peak rate (night) tariff. The primary energy input to the solar collector is direct solar energy which is free source of energy. The environmental impact of using each fuel is also indicated in terms of the rate of CO_2 emissions. The rate of CO_2 emissions of a fuel is strongly depended on its carbon content. The UK Currently taxed on a carbon equivalent basis at a rate for 2019 of £18 per tonne [182].

Table 6.4 Fuel Price and CO_2 emissions for each heating technology [183, 184]

Heating technology	Primary fuel type	Fuel price	Lower Heating Value (LHV)	CO_2 emissions
Solar collector	Solar energy	-	-	0 (kg/kWh)
Gas boiler	Gas	0.0363(£/kWh)	50 MJ/kg (13.9 kWh/kg)	0.184 (kg/kWh)
Biomass boiler	Wood chip (30% moisture content)	0.11(£/kg)	12.180 MJ/kg (3.38 kWh/kg) [185]	0.019 (kg/kWh)
	Wood pellets	0.24(£/kg)	17.94 MJ/kg (4.98kwh/kg) [186]	0.016 (kg/kWh)
Heat pump	Electricity Off -peak rate	0.0808(£/kWh)	-	0.241 (kg/kWh)
	Standard rate	0.1433(£/kWh)		

From the simulation model in Chapter 5, the start-up frequency of the heat generating system (heat pump, biomass, and gas boiler) are given In Table 6.5.

Table 6.5 The number of start-up times of the heating systems

	Heat pump	Biomass boiler	Gas boiler
Scenario 1	808	935	90
Scenario 2	965	784	175
Scenario 3	946	875	186

The cost of electricity of heat generating system (heat pump, biomass, and gas boiler) can be divided into standard rate (day) tariff and off-peak (night) tariff. The running hours for heat generations during day time and night time are given in Table 6.6 shows.

Table 6.6 The day time and night time of the heating systems

Design scenario	Schedule	Heat pump (h)	Biomass boiler (h)	Gas boiler (h)
Scenario 1	Day	411	1530	84
	Night	780	245	6
Scenario 2	Day	660	1854	134
	Night	838	253	46
Scenario 3	Day	601	1749	138
	Night	830	249	49

As the maximum heat demand of the site was 44 kW and the designed supply/return temperature is 65 °C/45 °C, the maximum of mass flow rate of water was $1.516m^3/h$ according Equation (6-17). The manufacture data of the water pump is given in Table 6.7.

Table 6.7 The manufactural data for heat pump

Equipment	Maximus of mass flow rate (m^3/h)	Maximum operating pressure (Bar)	The maximum power (W)
Water pump	2.9	10	45

6.4 Results and data analysis

6.4.1 Economic analysis

The cost of each heat generation system of the LTDH was evaluated according to the calculation procedures outlined in previous section, including the cost of fuel, electricity, maintenance, taxation for carbon emission and government subsidy. These costs are aggregated by design scenario as shown in Table 6.8 to 6.10.

Table 6.8 Operation cost for each element in system for Scenario 1.

Scenario 1	Fuel cost (£)	Electricity cost (£)	Maintenance (£)	Subsidy (£)	Taxation for Carbon emission (£)	Total operation cost (£)
Biomass boiler	1190.80	17.45	100.00	-276.63	9.11	1040.73
Gas boiler	71.87	0.21	275.00	0.00	5.96	353.04
Heat pump	------	338.94	360.00	-231.31	14.35	481.98
Thermal storage	------	------	105.65	0.00	0.00	105.65
Water pump	------	49.30	------	0.00	------	49.30
Solar collector	------	------	65.00	-97.83	0.00	-32.83

From Table 6.8, biomass boiler had the highest total operation cost amongst all other heat generation with annual cost of £1040.73. The solar collector with no primary fuel cost benefited from government subsidy and when the cost maintenance was subtracted it produced an annual income of £ 32.83. The heat pump was the second largest contributor of total operation cost to the LTDH with annual operation cost of £481.98. Gas boiler and thermal store ranked third and fourth with operation cost of £ 53.04 and £105.65, while the annual cost of water pump was only £49.30.

Similar analytical procedure was repeated for scenario 2, the results of the analysis are recapitulated in Table 6.9. It can be seen that similar ranking order of operation cost of the heat generating system is observed compared to scenario 1. The annual operation cost of biomass boiler decreased slightly to £1038.54, while annual operation cost of heat pump and gas boiler increased significantly to £538.27 and £431.01 respectively. Because of the increased heat capacity of solar collector, the annual income increased to £100.65

Table 6.9 Operation cost for each element in system for Scenario 2.

Scenario 2	Fuel cost (£)	Electricity cost (£)	Maintenance cost (£)	Subsidy (£)	Taxation for Carbon emission (£)	Total operation cost (£)
Biomass boiler	1132.51	18.88	100.00	-218.92	6.07	1038.54
Gas boiler	143.70	0.39	275.00	0.00	11.92	431.01
Heat pump	------	451.16	360.00	-290.94	18.05	538.27
Thermal storage	------	------	94.84	0.00	0.00	94.84
Water pump	------	49.30	------	0.00	------	49.30
Solar collector	------	------	95.00	-195.65	0.00	-100.65

The same cost trend was also seen in scenario 3, where the results of the analysis are recapitulated in Table 6.10. The annual operation cost of biomass boiler and heat pump decreased to £ 1003.02 and £ 525.17 respectively, while annual operation cost of gas boiler increased significantly to £ 477.14 respectively. The total annual income from the operation of the solar collector increased further to £ 168.48.

Table 6.10 Operation cost for each element in system for Scenario 3

Scenario 3	Fuel cost (£)	Electricity cost (£)	Maintenance cost (£)	Subsidy (£)	Taxation for Carbon emission (£)	Total operation cost (£)
Biomass boiler	1085.91	17.87	100.00	-207.59	6.83	1003.02
Gas boiler	186.21	0.45	275.00	0.00	15.48	477.14
Heat pump	------	425.86	360.00	-277.93	17.24	525.17
Thermal storage	------	------	117.62	0.00	0.00	117.62
Water pump	------	49.30	------	0.00	------	49.30
Solar collector	------	------	125.00	-293.48	0.00	-168.48

From the cost analysis given in Table 6.8 to 6.10, the total system operation cost for each scenario were illustrated in Figure 6.4. It can be seen that difference between annual cost of the three scenarios is small. The lowest system operation cost is scenario 1 with a total annual cost of £ 1997.87 while scenario 2 recorded the highest cost at £ 2051.31, a difference of only 2.67%.

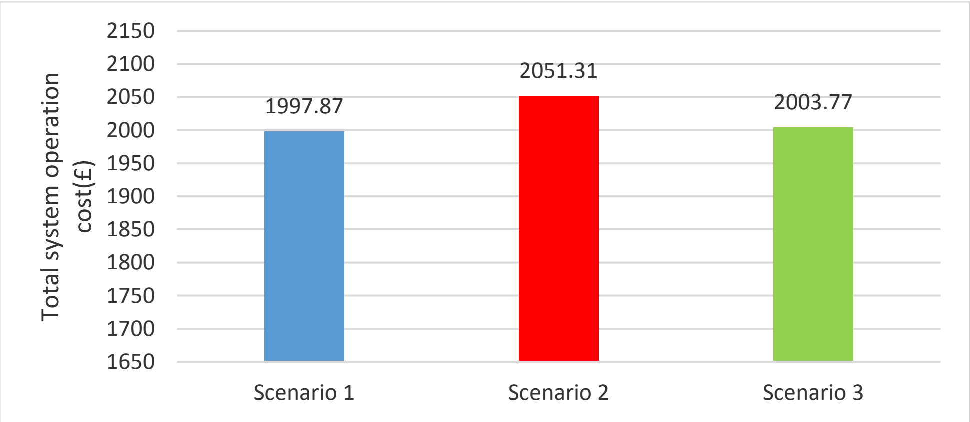


Figure 6.1 Total system operation cost for each scenario

6.4.2 Environmental analysis

The environmental analysis mainly focused on the carbon emission, which is based on a series of emission factors commonly accepted for each of energy sources, as shown in Table 6.4. Using the total heat energy generated by each heat source given in Table 6.1, the carbon emission for each heat source can be calculated for the three scenarios.

For scenario 1, the heat pump was the largest carbon emission dioxide emitter and produced 797.3 kg of CO_2, which accounted for 48.78% of total carbon emissions as Figure 6.5 shows. The biomass boiler was the second largest of total carbon emission with 505.875kg, which represented 30.95% of total emission. In the third place the gas boiler was with 20.26% of total emission. The large proportion of emission from heat pump is related to emission factor of the grid (0.241 kg/kWh) which is larger than burning biomass (0.019 kg/kWh) or gas (0.184 kg/kWh) and also to the number of running hours per year.

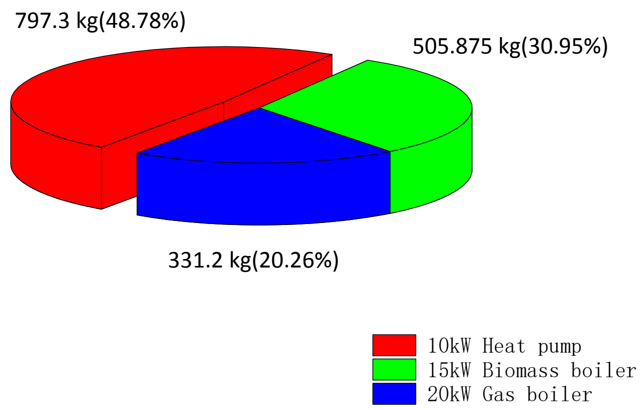


Figure 6.2 Carbon emission for each heat source in Scenario 1.

For scenario 2, the heat pump also contributed the highest amount of carbon emission with a proportion of 50.08% (1002.83 kg) of total carbon emission as Figure 6.3 shows. In this scenario, however, the biomass boiler emitted less carbon (337.12 kg) compared to the gas boiler (662.4 kg) with a proportion of 5.03% and 9.88% respectively.

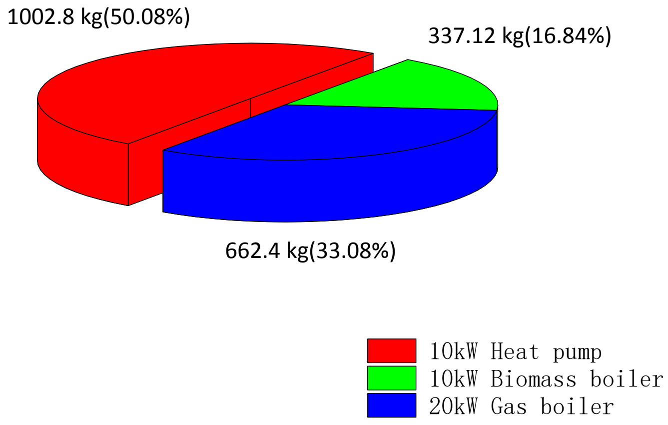


Figure 6.3 Carbon emission for each heat source in Scenario 2

For scenario 3, the quantities of carbon emission are still larger than those emitted in scenario 1 and 2. The largest carbon emission was still produced by heat pump, with a total of 957.98kg or 43.59% of total emission, as Figure 6.4 shows. Gas boiler ranked second largest carbon emission, accounting for 39.14% of total system carbon emission. This was because in this scenario the total heat demand supplied by the gas boiler accounted for 11.21%, which lead to the increase of gas boiler carbon emission. The biomass boiler was the lowest share of carbon emission with 379.62 kg.

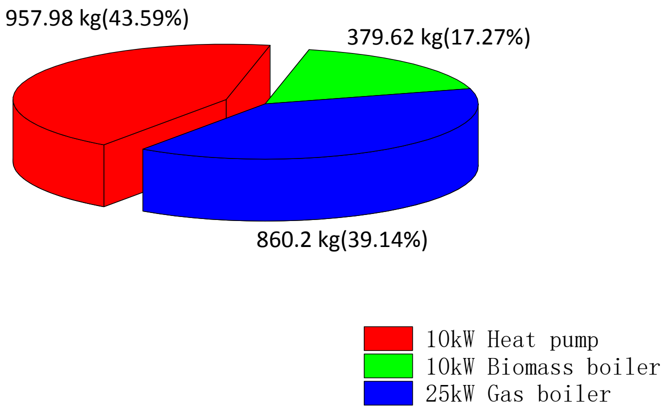


Figure 6.4 Carbon emission for each heat source in Scenario 3.

Overall, in all design scenario, the heat pump emits the highest proportion of total carbon emission of the site. The low emission factor of biomass boiler fuel is translated to low carbon emission of the biomass boiler (less than 40% of total carbon emission) even though it generates about 50% of total heat demand. This represents the lowest emission besides the solar collector. The total carbon emission for generating heat to the site given by the three design is shown in Figure 6.5. The lowest carbon emission was scenario 1 with 1634.4 kg, which represents a saving of 25.63% compared with the highest system carbon emission of scenario 3.

The carbon emission of each technology is based on current emission factors. In the future when the electricity grid is decarbonised such as hydropower and wind power, the carbon emission and energy consumption from the LTDH system can be largely lower and realise the objective of zero carbon emission.

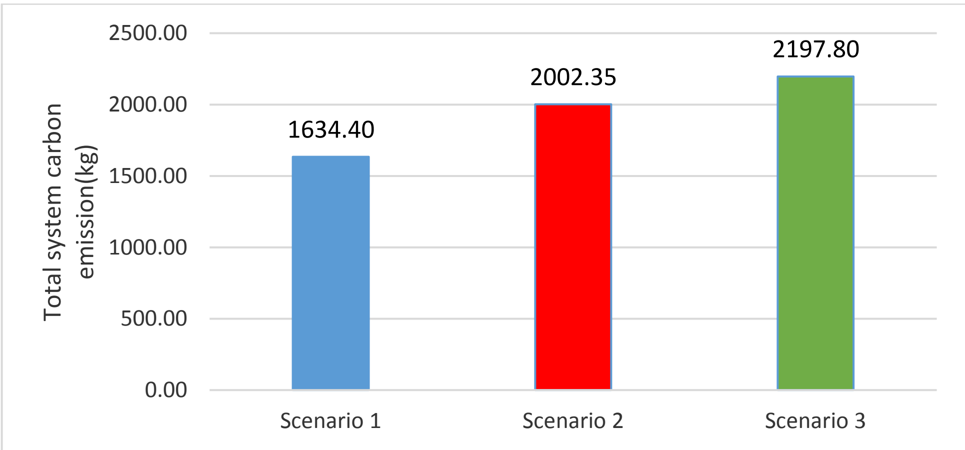


Figure 6.5 Total system carbon emission at different scenarios.

6.5 Summary

In this chapter the cost of generating heat and the CO_2 emission resulting from the process was quantified. The cost of generating heat from multiple heat sources for the LTDH system considered fuel cost, maintenance and government subsidies.

It was shown that the annual operation cost of biomass boiler was the largest for three scenarios. The solar collector on the other hand generated an income due to government subsidies and free incident solar energy (primary energy) and the level of income increased with the size of solar collector. The heat pump which ran of the electricity grid had the second largest annual operation cost. The heat generation of design scenario 1 resulted in the lowest system operation cost of £ 1997.87 compared to the scenario 2 and 3 of £ 2051.31 and £ 2003.77 respectively. Overall, there is little difference in terms of annual operation cost between the three cases.

Furthermore, in terms of carbon dioxide emission, the heat pump contributed the largest proportion (higher than 40%) in the three scenarios. The remaining emission was shared between biomass boiler and the gas boiler. As the carbon dioxide emission factor of biomass fuel is a fold lower than that of natural gas, the biomass boiler emission accounted for less than 40% of the total emission, even though it supplied about 50% of the heat demand of the site. The lowest emission of generating heat was obtained with design scenario 1 of 1634.4 kg while scenario 3 recorded the highest emission of 2197.8 kg. Overall, scenario 1 with a combination of 1.56 kW solar collector, 10 kW heat pump, 15 kW biomass boiler, 20 kW gas boiler and a thermal store of 0.894 m^3 (or 25.96 kWh) produced more favourable economic and environmental impact.

7 Conclusions, discussions and future work

7.1 Introduction

Heat provision in buildings is commonly achieved through individual fossil fuel boilers or renewable heat sources (heat pump, biomass and solar energy). District heating technology has been successfully deployed in many countries to provide thermal comfort in buildings. Energy saving and emission reduction are becoming the focus in many countries. With the advancement in smart cities design, networked services are increasingly sought. Through the heat network conception is not a new idea, the provision of low temperature heat through a LTDH network is increasingly promoted as an effective solution to increase efficiency and adoption of renewable energy systems. Integrating multiple heat sources into a single heat network is also a novel method of providing heat supply flexibility and diversity. In this work the main aim and objective were addressed through reviewing an extended library of published articles, defining the economic and environmental viability of the system.

Therefore, this chapter outlines the research project conclusion, discussion on the trends of the research field, contribution, limitation and further development work.

7.2 Contribution

This work sought to advance knowledge of design of LTDH using multiple heat sources as a way to mitigate climate change and conserve energy resources. The research work particularly contributed to the following:

i. A literature review of district heating network based on previous published articles, studies, as well as books and official sites in this field.

ii. A simulation of the dynamic heat demand of the Creative Energy Homes (CEHs) using EnergyPlus software to aggregate the site’s heat demand.

iii. Development of a mathematical formulation of optimum heat sources capacities in a multiple heat sources LTDH.

iv. Development of an optimisation of the thermal store to satisfy the site's heat load reliability.

v. Quantification of the operational cost and environmental impact of the multiple heat generating sources in a LTDH system.

7.2.1 Review of DH technology and its role in the future heating market

The literature review on DH technology covered the main technologies in DH system including heat sources, heat networks, supply and thermal storage, a detailed description of the evolution of traditional DH technology and projected future LTDH systems with the flexibility to make use of low-cost heat sources, heat recovery and renewable sources, leading the transition towards a low carbon economy. This particularly focused on giving an insight into the actual heating sector worldwide.

7.2.2 Optimisation of the heat sources and thermal store in LTDH

The heat generating technologies were selected from currently available technologies that can supply heat to the LTDH and satisfy the heat demand of the site. Three scenarios with variable technologies and sizes were considered. The contribution of renewable heat energy from the solar collector was maximised determining the optimum thermal store capacity that can accommodate all intermittent heat generated from the solar collector. The methodology of defining the optimal heat sources and thermal store capacity was carried out to meet the hourly load profile of the site for a year. A reliability analysis was then executed to ensure the site's heat demand met for 100% of the year.

7.2.3 Economic and environmental analysis

The economic and environmental viability of the heat generating system was determined through a simplified formulation of the annual operating cost and carbon emission.

This considered the fuel tariffs, maintenance, government subsidies and carbon intensity of the fossil fuels and power grid.

Therefore, this work has sought to advance both the concept of LTDH and integration of multiple heat generating sources for the provision of low carbon heat in a small-scale community heating scheme. This was achieved through mathematical formulation of the basic and fundamental theory and computer simulation for sizing and optimizing the heat sources and thermal capacity.

7.3 Conclusion

In this thesis, the main key finding and contribution of this work are related to the heat load of CEHs, optimise multiple heat sources and thermal storage, and LTDH system carbon emission and operation cost.

7.3.1 Community energy demand

The aggregate load of the Creative Energy Homes was 44 kW, while the annual energy consumption was 40258.1 kWh, including 14110.89 kWh for domestic hot water and 26417.92 kWh for space heating. Except two peaks of heat load profiles in morning and evening, weekend heat load profile had one more peak point than weekdays profile. The CEHs duration curve was obtained, the heat load from 15 kW to 44 kW was considered in the peak demand bracket, which accounted for 33.08% of the year.

7.3.2 Optimisation of multiple heat sources and thermal store

The biomass boiler and heat pump supplied more than 80% of heat demand of the site, while the gas boiler fulfilled less than 10% of heat demand, working as auxiliary boiler. The solar collector operated for a total of 1891 hours per year and contributed less than 10% of heat demand. Excluding the running time of solar collector, the biomass boiler was the longest operation device, while the running time of heat pump and gas boiler ranked second and third.

As the heat capacity of solar collector increased, the heat loss of thermal store increased. The thermal store heat loss with 4.67 kW solar collector increased by 46.64% compared with 1.56 kW solar collector.

As the thermal store heat loss is considered into dynamic analysis, the running time of main heat generations increased and the maximum thermal store capacity decreased during a year, which can help to prevent the selection of oversized thermal store.

7.3.3 Economic and environmental analysis in LTDH

The annual operation cost of biomass boiler was the largest in all heat generations, while it contributed less than 10% of total carbon emission and 50% of heat demand. However, the heat pump had the largest annual operation cost with largest proportion (higher than 80%) in carbon emission. The solar collector generated an income and the level of income increased with the size of solar collector.

The system consisted a 1.56 kW solar collector, 10 kW heat pump, 15 kW biomass boiler, 20 kW gas boiler and a thermal store of 0.894 m^3 (or 25.96 kWh) which can supply heat for the site in a LTDH system at the lowest cost and with the least environment impact. The system's annual operation cost and carbon emission was £ 1997.87and 1634.4 kg respectively.

7.3.4 Limitation

Despite a concerted effort to address the many challenges of the project brief, the scope of the project could be enhanced further. Some limitations are related to access expensive commercial software packages that can enhance the results of the study. Another limitation of the work is to conduct a validation analysis of the simulation data using the LTDH network of the Creative Energy Homes. This however will require full access to the dwellings and observe ethical issues of residents' data. Finally, this project sought to help the adoption heat generation from renewable heat sources only and the power generation from renewable sources was not considered.

7.4 Future work

The results of this work can form the basis for future development of LTDH as a low carbon heat supply for an agglomeration of buildings. Future works could expend on the present study to include other element of the heat network such as heat distribution system, heat interface units and low temperature heat emission radiators on the end user side. The heat distribution would take into account pressure, flow velocity and heat loss in the thermally insulated pipework. Importantly, developing novel methods of integrated an increasingly larger share of heat supplied from renewable sources and consideration of the provision of domestic hot water.

References

1. Dudley, B. *BP Statistical Review of World Energy*. in *World Petroleum Congress: London*. 2019.
2. IEA. *CO2 emissions from fuel combustion*. 2018; Available from: https://webstore.iea.org/download/direct/2373?fileName=CO2_Emissions_from_Fuel_Combustion_2018_Highlights.pdf.
3. ICPP. *Global Warming of 1.5°C: Summary for Policymakers*. 2018; Available from: https://library.wmo.int/doc_num.php?explnum_id=9954.
4. League, E., P. Kabat, P. Egerton, O. Baddour, L. Paterson, C. Nullis, S. Castonguay, and M. Walsh, *United in Science: High-level Synthesis Report of Latest Climate Science Information convened by the Science Advisory Group of the UN Climate Action Summit 2019.* 2019.
5. IEA. *Energy Efficiency: Buildings*. 2019; Available from: https://www.iea.org/topics/energyefficiency/buildings/.
6. GOV.UK. *Energy Consumption in the UK(ECUK) 2018*. 2019; Available from: https://www.gov.uk/government/statistics/energy-consumption-in-the-uk.
7. Business, D.f. *Energy and Industrial Strategy (2017) Clean Growth Strategy*. 2018; Available from: https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/700496/clean-growth-strategy-correction-april-2018.pdf.
8. ICAX, *Uncomfortable Home Truths: Why Britain urgently needs a low carbon strategy.* 2019.
9. Department for Business, E.a.I.S. *Final UK greenhouse gas emissions national statistics*. 2019; Available from: https://data.gov.uk/dataset/9568363e-57e5-4c33-9e00-31dc528fcc5a/final-uk-greenhouse-gas-emissions-national-statistics.
10. Act, C.C. *The target for 2050*. 2008; Available from: http://www.legislation.gov.uk/ukpga/2008/27/section/1.
11. council, U.G.B. *Buildings- Committee on Climated Change*. 2015; Available from: https://www.theccc.org.uk/wp-content/uploads/2014/08/Fact-sheet-buildings-updated-July-2015.pdf.
12. Trust, E.S. *The Clean Growth Plan: A "2050-ready" new-build homes policy*. 2017; Available from: https://www.energysavingtrust.org.uk/sites/default/files/reports/ERP4_The%20Clean%20Growth%20Plan_A%202050-ready%20new-build%20homes%20policy.pdf.
13. IET. *Scaling Up Retrofit 2050*. 2019; Available from: https://www.theiet.org/impact-society/factfiles/built-environment/retrofit-2050/?utm_source=redirect&utm_medium=legacyredirects&utm_campaign=2019relaunch.
14. strategy, D.f.B.E.I. *2017 UK GREENHOUSE GAS EMISSIONS, PROVISIONAL FIGURES*. 2017.
15. Yang, X., H. Li, and S. Svendsen, *Decentralized substations for low-temperature district heating with no Legionella risk, and low return temperatures.* Energy, 2016. **110**: p. 65-74.
16. Di Lucia, L. and K. Ericsson, *Low-carbon district heating in Sweden – Examining a successful energy transition.* Energy Research & Social Science, 2014. **4**: p. 10-20.
17. Skagestad, B. and P. Mildenstein, *District heating and cooling connection handbook*. 2002: NOVEM, Netherlands Agency for Energy and the Environment.
18. Delangle, A., R.S. Lambert, N. Shah, S. Acha, and C.N. Markides, *Modelling and optimising the marginal expansion of an existing district heating network.* Energy, 2017. **140**: p. 209-223.
19. Beith, R., *Small and micro combined heat and power (CHP) systems: advanced design, performance, materials and applications*. 2011: Elsevier.
20. Boukhanouf, R., *Small combined heat and power (CHP) systems for commercial buildings and institutions.* Teoksessa Beith, R.(toim.), 2011: p. 365-394.

21. Darrow, K., R. Tidball, J. Wang, and A. Hampson, *Catalog of CHP technologies.* US Environmental Protection Agency, Washington, DC, 2015.
22. Yu, Z., J. Han, X. Cao, W. Chen, and B. Zhang, *Analysis of total energy system based on solid oxide fuel cell for combined cooling and power applications.* International Journal of Hydrogen Energy, 2010. **35**(7): p. 2703-2707.
23. Chua, K.J. and S.K. Chou, *A modular approach to study the performance of a two-stage heat pump system for drying.* Applied Thermal Engineering, 2005. **25**(8–9): p. 1363-1379.
24. Park, K.-J. and D. Jung, *Performance of heat pumps charged with R170/R290 mixture.* Applied energy, 2009. **86**(12): p. 2598-2603.
25. Chow, T.T., G. Pei, K. Fong, Z. Lin, A. Chan, and M. He, *Modeling and application of direct-expansion solar-assisted heat pump for water heating in subtropical Hong Kong.* Applied Energy, 2010. **87**(2): p. 643-649.
26. Zhang, Y., Q. Ma, B. Li, X. Fan, and Z. Fu, *Application of an air source heat pump (ASHP) for heating in Harbin, the coldest provincial capital of China.* Energy and Buildings, 2017. **138**: p. 96-103.
27. Ding, Y., G. Ma, Q. Chai, and Y. Jiang, *Experiment investigation of reverse cycle defrosting methods on air source heat pump with TXV as the throttle regulator.* International Journal of Refrigeration, 2004. **27**(6): p. 671-678.
28. Jiang, Y., H. Fu, Y. Yao, L. Yan, and Q. Gao, *Experimental study on concentration change of spray solution used for a novel non-frosting air source heat pump system.* Energy and Buildings, 2014. **68, Part B**: p. 707-712.
29. Wang, F., Z. Wang, Y. Zheng, Z. Lin, P. Hao, C. Huan, and T. Wang, *Performance investigation of a novel frost-free air-source heat pump water heater combined with energy storage and dehumidification.* Applied Energy, 2015. **139**: p. 212-219.
30. Lucia, U., M. Simonetti, G. Chiesa, and G. Grisolia, *Ground-source pump system for heating and cooling: Review and thermodynamic approach.* Renewable and Sustainable Energy Reviews.
31. Chua, K., S. Chou, and W. Yang, *Advances in heat pump systems: A review.* Applied Energy, 2010. **87**(12): p. 3611-3624.
32. De Pasquale, A.M., A. Giostri, M.C. Romano, P. Chiesa, T. Demeco, and S. Tani, *District heating by drinking water heat pump: Modelling and energy analysis of a case study in the city of Milan.* Energy, 2017. **118**: p. 246-263.
33. Zamfirescu, C. and I. Dincer, *Performance investigation of high-temperature heat pumps with various BZT working fluids.* Thermochimica Acta, 2009. **488**(1–2): p. 66-77.
34. Yang, W., J. Zhou, W. Xu, and G. Zhang, *Current status of ground-source heat pumps in China.* Energy Policy, 2010. **38**(1): p. 323-332.
35. Ni, L., J. Dong, Y. Yao, C. Shen, D. Qv, and X. Zhang, *A review of heat pump systems for heating and cooling of buildings in China in the last decade.* Renewable Energy, 2015. **84**: p. 30-45.
36. Li, H., L. Sun, and Y. Zhang, *Performance investigation of a combined solar thermal heat pump heating system.* Applied Thermal Engineering, 2014. **71**(1): p. 460-468.
37. Dai, L., S. Li, L. DuanMu, X. Li, Y. Shang, and M. Dong, *Experimental performance analysis of a solar assisted ground source heat pump system under different heating operation modes.* Applied Thermal Engineering, 2015. **75**: p. 325-333.
38. Ma, H., C. Li, W. Lu, Z. Zhang, S. Yu, and N. Du, *Investigation on a solar-groundwater heat pump unit associated with radiant floor heating.* Renewable and Sustainable Energy Reviews.
39. Wang, W., Z. Ma, Y. Jiang, Y. Yang, S. Xu, and Z. Yang, *Field test investigation of a double-stage coupled heat pumps heating system for cold regions.* International Journal of Refrigeration, 2005. **28**(5): p. 672-679.

40. Lund, R., D.D. Ilic, and L. Trygg, *Socioeconomic potential for introducing large-scale heat pumps in district heating in Denmark.* Journal of Cleaner Production, 2016. **139**: p. 219-229.
41. Olsen, P., H. Lambertsen, R. Hummelshøj, B. Bøhm, C. Christiansen, S. Svendsen, C. Larsen, and J. Worm. *A new low-temperature district heating system for low-energy buildings*. in *Proceedings of the 11th International Symposium on District Heating and Cooling, Iceland*. 2008.
42. Rad, F.M. and A.S. Fung, *Solar community heating and cooling system with borehole thermal energy storage – Review of systems.* Renewable and Sustainable Energy Reviews, 2016. **60**: p. 1550-1561.
43. Zhai, X.Q., J.R. Yang, and R.Z. Wang, *Design and performance of the solar-powered floor heating system in a green building.* Renewable Energy, 2009. **34**(7): p. 1700-1708.
44. Zhai, X.Q. and R.Z. Wang, *Experiences on solar heating and cooling in China.* Renewable and Sustainable Energy Reviews, 2008. **12**(4): p. 1110-1128.
45. Zhai, H., Y.J. Dai, J.Y. Wu, and R.Z. Wang, *Energy and exergy analyses on a novel hybrid solar heating, cooling and power generation system for remote areas.* Applied Energy, 2009. **86**(9): p. 1395-1404.
46. Zhao, D., J. Ji, H. Yu, W. Wei, and H. Zheng, *Numerical and experimental study of a combined solar Chinese kang and solar air heating system based on Qinghai demonstration building.* Energy and Buildings, 2017. **143**: p. 61-70.
47. Bauer, D., R. Marx, and H. Drück, *Solar District Heating Systems for Small Districts with Medium Scale Seasonal Thermal Energy Stores.* Energy Procedia, 2016. **91**: p. 537-545.
48. Verma, V. and K. Murugesan, *Experimental study of solar energy storage and space heating using solar assisted ground source heat pump system for Indian climatic conditions.* Energy and Buildings, 2017. **139**: p. 569-577.
49. Lizana, J., C. Ortiz, V.M. Soltero, and R. Chacartegui, *District heating systems based on low-carbon energy technologies in Mediterranean areas.* Energy.
50. IRENA. *Renewable energy in district heating and cooling: A sector roadmap for Remap*. 2019; Available from: https://www.irena.org/publications/2017/Mar/Renewable-energy-in-district-heating-and-cooling.
51. Saxena, R.C., D.K. Adhikari, and H.B. Goyal, *Biomass-based energy fuel through biochemical routes: A review.* Renewable and Sustainable Energy Reviews, 2009. **13**(1): p. 167-178.
52. Ericsson, K. and S. Werner, *The introduction and expansion of biomass use in Swedish district heating systems.* Biomass and Bioenergy, 2016. **94**: p. 57-65.
53. Verma, V.K., S. Bram, and J. De Ruyck, *Small scale biomass heating systems: Standards, quality labelling and market driving factors – An EU outlook.* Biomass and Bioenergy, 2009. **33**(10): p. 1393-1402.
54. Fiedler, F., *The state of the art of small-scale pellet-based heating systems and relevant regulations in Sweden, Austria and Germany.* Renewable and Sustainable Energy Reviews, 2004. **8**(3): p. 201-221.
55. Sartor, K., S. Quoilin, and P. Dewallef, *Simulation and optimization of a CHP biomass plant and district heating network.* Applied Energy, 2014. **130**: p. 474-483.
56. Noussan, M., G. Cerino Abdin, A. Poggio, and R. Roberto, *Biomass-fired CHP and heat storage system simulations in existing district heating systems.* Applied Thermal Engineering, 2014. **71**(2): p. 729-735.
57. Chasapis, D., V. Drosou, I. Papamechael, A. Aidonis, and R. Blanchard, *Monitoring and operational results of a hybrid solar-biomass heating system.* Renewable Energy, 2008. **33**(8): p. 1759-1767.
58. Carlon, E., M. Schwarz, L. Golicza, V.K. Verma, A. Prada, M. Baratieri, W. Haslinger, and C. Schmidl, *Efficiency and operational behaviour of small-scale pellet boilers installed in residential buildings.* Applied energy, 2015. **155**: p. 854-865.

59. Tereshchenko, T. and N. Nord, *Importance of Increased Knowledge on Reliability of District Heating Pipes.* Procedia Engineering, 2016. **146**: p. 415-423.

60. Smidt, H.D., Hansen, K.E., *Stabiliser consumption in HDPE casing pipes*. 2006, Danish District Heating Association.

61. Thörnblom, K., S. Forsaeus Nilsson, S. Sällberg, G. Bergström, C. Ek, and A. Stenström, *Durability of non-pressure polypropylene pipe materials.* Report/SP Technical Research Institute of Sweden, 2007. **30**.

62. Nilsson, S., K. Thörnblom, S. Sällberg, and G. Bergström. *Brittle fracture in casing pipes*. in *Proc. 10th international symposium on district heating and cooling, Hanover, Germany*. 2006.

63. Mangs, S., M. Fröling, O. Ramnäs, and U. Jarfelt, *Transport of 1, 1, 1, 3, 3-pentafluorobutane (HFC-365mfc) in rigid polyurethane foam and polyethylene.* Cellular polymers, 2002. **21**(3): p. 155-164.

64. Ramnäs, O. and U. Jarfelt, *New materials and constructions for improving the quality and lifetime of district heating pipes including joints–thermal, mechanical and environmental performance.* District Heating and Cooling report 8DHC-08.01, 2008.

65. Smidt, H.D.H., K.E., *Stabiliser consumption in HDPE casing pipes.* Danish District Heating Association, 2006. **02**.

66. Ramnäs, O., Jarfelt, U.,, *New materials and constructions for improving the quality and lifetime of district heating pipes including joints – thermal, mechanical and environmental performance,*. 2008, IEA District Heating and Cooling.

67. Zinko, H., B. Bøhm, H. Kristjansson, U. Ottosson, M. Rama, and K. Sipila, *District heating distribution in areas with low heat demand density.* International Energy Agency, 2008.

68. Korsman, J., S. De Boer, and I. Smits, *Cost benefits and long term behaviour of a new all plastic piping system.* District Heating and Cooling report 8DHC-08-05, 2008.

69. Berge, A., B. Adl-Zarrabi, and C.-E. Hagentoft, *Assessing the Thermal Performance of District Heating Twin Pipes with Vacuum Insulation Panels.* Energy Procedia, 2015. **78**: p. 382-387.

70. Berge, A. and B. Adl-Zarrabi. *Evaluation of vacuum insulation panels used in hybrid insulation district heating pipes*. in *Proceedings of the 14th international symposium on district heating and cooling, Stockholm, Sweden*. 2014.

71. Bøhm, B. and H. Kristjansson, *Single, twin and triple buried heating pipes: on potential savings in heat losses and costs.* International journal of energy research, 2005. **29**(14): p. 1301-1312.

72. BOYLE, K. *DISTRICT HEATING & HEAT NETWORK DESIGN* 2014; Available from: http://www.usewoodfuel.co.uk/media/411085/k._boyle-district_heating_and_heat_network_design_bridge_of_allan_11_mar_2014.pdf.

73. Gut, J.A.W. and J.M. Pinto, *Modeling of plate heat exchangers with generalized configurations.* International Journal of Heat and Mass Transfer, 2003. **46**(14): p. 2571-2585.

74. Abou Elmaaty, T.M., A.E. Kabeel, and M. Mahgoub, *Corrugated plate heat exchanger review.* Renewable and Sustainable Energy Reviews, 2017. **70**: p. 852-860.

75. Installations, E.T.F.C., *Guidelines for District Heating Substations.* Euroheat & Power, 2008: p. 68.

76. Sable, M.J., *Analysis of Natural Convention Heat Transfer Enhancement in Finned Tube Heat Exchangers.*

77. Chen, T., J. Wang, and W. Peng, *Flow and heat transfer analyses of a plate-fin heat exchanger in an HTGR.* Annals of Nuclear Energy, 2017. **108**: p. 316-328.

78. Fumey, B., R. Weber, and L. Baldini, *Liquid sorption heat storage – A proof of concept based on lab measurements with a novel spiral fined heat and mass exchanger design.* Applied Energy, 2017. **200**: p. 215-225.

79. Menéndez-Díaz, A., C. Ordóñez-Galán, J.B. Bouza-Rodríguez, and J.J. Fernández-Calleja, *Thermal analysis of a stoneware panel covering radiators.* Applied Energy, 2014. **131**: p. 248-256.
80. Zhou, G. and J. He, *Thermal performance of a radiant floor heating system with different heat storage materials and heating pipes.* Applied Energy, 2015. **138**: p. 648-660.
81. Price, M., K. Butcher, and E. Chartered Institution of Building Services, *Fan coil units.* 2008.
82. Wang, Y., Z. Zhang, F. He, and X. Liu, *Experimental Investigations on Thermal Performance of a New Floor Heating Device.* Procedia Engineering, 2015. **121**: p. 2132-2140.
83. Atienza Márquez, A., J.M. Cejudo López, F. Fernández Hernández, F. Domínguez Muñoz, and A. Carrillo Andrés, *A comparison of heating terminal units: Fan-coil versus radiant floor, and the combination of both.* Energy and Buildings, 2017. **138**: p. 621-629.
84. Myhren, J.A. and S. Holmberg, *Flow patterns and thermal comfort in a room with panel, floor and wall heating.* Energy and Buildings, 2008. **40**(4): p. 524-536.
85. Hasan, A., J. Kurnitski, and K. Jokiranta, *A combined low temperature water heating system consisting of radiators and floor heating.* Energy and Buildings, 2009. **41**(5): p. 470-479.
86. Olesen, B.W., *Radiant floor heating in theory and practice.* ASHRAE journal, 2002. **44**(7): p. 19.
87. Hauer, A. and Z. Bayern. *Storage Technology Issues and Opportunities*. in *Committee on Energy Research and Technology (International Energy Agency), International Low-Carbon Energy Technology Platform, Strategic and Cross-Cutting Workshop Energy Storage–Issues and Opportunities*. 2011.
88. Kalaiselvam, S. and R. Parameshwaran, *Thermal Energy Storage Technologies for Sustainability: Systems Design, Assessment and Applications*. 2014: Elsevier.
89. Lund, H., S. Werner, R. Wiltshire, S. Svendsen, J.E. Thorsen, F. Hvelplund, and B.V. Mathiesen, *4th Generation District Heating (4GDH): Integrating smart thermal grids into future sustainable energy systems.* Energy, 2014. **68**: p. 1-11.
90. Lake, A., B. Rezaie, and S. Beyerlein, *Review of district heating and cooling systems for a sustainable future.* Renewable and Sustainable Energy Reviews, 2017. **67**: p. 417-425.
91. Sayegh, M.A., J. Danielewicz, T. Nannou, M. Miniewicz, P. Jadwiszczak, K. Piekarska, and H. Jouhara, *Trends of European research and development in district heating technologies.* Renewable and Sustainable Energy Reviews, 2017. **68**: p. 1183-1192.
92. Mazhar, A.R., S. Liu, and A. Shukla, *A state of art review on the district heating systems.* Renewable and Sustainable Energy Reviews, 2018. **96**: p. 420-439.
93. Xiong, W., Y. Wang, B.V. Mathiesen, H. Lund, and X. Zhang, *Heat roadmap China: new heat strategy to reduce energy consumption towards 2030.* Energy, 2015. **81**: p. 274-285.
94. Nuorkivi, A., *2 - District heating and cooling policies worldwide A2 - Wiltshire, Robin*, in *Advanced District Heating and Cooling (DHC) Systems*. 2016, Woodhead Publishing: Oxford. p. 17-41.
95. Korppoo, A. and N. Korobova, *Modernizing residential heating in Russia: End-use practices, legal developments, and future prospects.* Energy Policy, 2012. **42**: p. 213-220.
96. Münster, M., P.E. Morthorst, H.V. Larsen, L. Bregnbæk, J. Werling, H.H. Lindboe, and H. Ravn, *The role of district heating in the future Danish energy system.* Energy, 2012. **48**(1): p. 47-55.
97. COUNTRY, C.B. *STATISTICS OVERVIEW*. 2016; Available from: https://www.euroheat.org/wp-content/uploads/2016/03/2015-Country-by-country-Statistics-Overview.pdf.
98. Wessberg, N., *Local decisions in the Finnish energy production network—a socio-technical perspective.* Landscape and urban planning, 2002. **61**(2-4): p. 137-146.
99. Woods, P. and J. Overgaard, *1 - Historical development of district heating and characteristics of a modern district heating system A2 - Wiltshire, Robin*, in *Advanced District Heating and Cooling (DHC) Systems*. 2016, Woodhead Publishing: Oxford. p. 3-15.

100. Deparment for Business, E.I.S. *Heat Nestworks Investment Project*. 2019; Available from: https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/691643/Heat_Network_Case_Study_Brochure.pdf.
101. Brand, M., A. Dalla Rosa, and S. Svendsen, *Performance of low-temperature district heating for low-energy houses.* 2010.
102. Dalla Rosa, A. and J.E. Christensen, *Low-energy district heating in energy-efficient building areas.* Energy, 2011. **36**(12): p. 6890-6899.
103. Østergaard, P.A. and H. Lund, *A renewable energy system in Frederikshavn using low-temperature geothermal energy for district heating.* Applied Energy, 2011. **88**(2): p. 479-487.
104. Zvingilaite, E., T.S. Ommen, B. Elmegaard, and M.L. Franck. *Low temperature district heating consumer unit with micro heat pump for domestic hot water preparation*. in *13th International Symposium on District Heating and Cooling*. 2012.
105. Fang, H., J. Xia, K. Zhu, Y. Su, and Y. Jiang, *Industrial waste heat utilization for low temperature district heating.* Energy Policy, 2013. **62**: p. 236-246.
106. Brand, M., J.E. Thorsen, and S. Svendsen, *Numerical modelling and experimental measurements for a low-temperature district heating substation for instantaneous preparation of DHW with respect to service pipes.* Energy, 2012. **41**(1): p. 392-400.
107. Paulsen, O., J. Fan, S. Furbo, and J.E. Thorsen. *Consumer unit for low energy district heating net*. in *The 11th International Symposium on district heating and cooling*. 2008.
108. Brand, M. and S. Svendsen, *Renewable-based low-temperature district heating for existing buildings in various stages of refurbishment.* Energy, 2013. **62**: p. 311-319.
109. China, I.O.o.t.S.C.T.P.s.R.o. *China's Policies and Actions for Addressing Climate Change*. 2011; Available from: http://www.scio.gov.cn/zfbps/ndhf/2011/Document/1052718/1052718.htm.
110. Department OF CLIMATE CHANGE , N.D.R.C.O.C. *China's INDC*. 2015; Available from: https://www4.unfccc.int/sites/submissions/INDC/Published%20Documents/China/1/China's%20INDC%20-%20on%2030%20June%202015.pdf.
111. Nekrasov, A., S. Voronina, and V. Semikashev, *Problems of residential heat supply in Russia.* Studies on Russian Economic Development, 2012. **23**.
112. RUSSIA, E.S.o. *ENERGY STRATEGY of RUSSIA FOR THE PERIOD UP TO 2030*. 2010; Available from: http://www.energystrategy.ru/projects/docs/ES-2030_(Eng).pdf.
113. Nekrasov, A., Y. Sinyak, and S. Voronina, *Development prospects for the heating supply in Russia.* Studies on Russian Economic Development, 2011. **22**: p. 130-141.
114. Paiho, S., R. Abdurafikov, H. Hoang, and J. Kuusisto, *An analysis of different business models for energy efficient renovation of residential districts in Russian cold regions.* Sustainable cities and society, 2015. **14**: p. 31-42.
115. Agency, U.S.E.P. *2012 joint report*. 2012; Available from: https://www.epa.gov/.
116. Union, E., *Directive 2009/28/EC of the European Parliament and of the Council of 23 April 2009 on the promotion of the use of energy from renewable sources and amending and subsequently repealing Directives 2001/77/EC and 2003/30/EC.* Official Journal of the European Union, 2009. **5**: p. 2009.
117. Lund, H., B. Möller, B.V. Mathiesen, and A. Dyrelund, *The role of district heating in future renewable energy systems.* Energy, 2010. **35**(3): p. 1381-1390.
118. Rinne, S. and S. Syri, *Heat pumps versus combined heat and power production as CO2 reduction measures in Finland.* Energy, 2013. **57**: p. 308-318.
119. Zaunbrecher, B.S., K. Arning, T. Falke, and M. Ziefle, *No pipes in my backyard?: Preferences for local district heating network design in Germany.* Energy Research & Social Science, 2016. **14**: p. 90-101.
120. Wissner, M., *Regulation of district-heating systems.* Utilities Policy, 2014. **31**: p. 63-73.
121. Wojdyga, K., M. Chorzelski, and E. Rozycka-Wronska, *Emission of pollutants in flue gases from Polish district heating sources.* Journal of cleaner production, 2014. **75**: p. 157-165.

122. Government, U. *Climate Change Act 2008*. 2008.
123. OFGEM. *Factsheet: The Renewable Heat Incentive Domestic or Non-Domestic?* 2016; Available from: https://www.ofgem.gov.uk/sites/default/files/docs/drhi_factsheet_therhidomornondom_v2_0_mar_2016_web.pdf.
124. OFGEM. *Heat demand limit*. 2018; Available from: https://www.ofgem.gov.uk/key-term-explained/heat-demand-limit.
125. OFGEM. *Non-Domestic RHI tariff rates*. 2018; Available from: https://www.ofgem.gov.uk/environmental-programmes/non-domestic-rhi/contacts-guidance-and-resources/tariffs-and-payments-non-domestic-rhi.
126. OFGEM. *Tariffs and payments: Domestic RHI*. 2018; Available from: https://www.ofgem.gov.uk/environmental-programmes/domestic-rhi/contacts-guidance-and-resources/tariffs-and-payments-domestic-rhi/current-future-tariffs.
127. *UK launches RHI scheme.* Renewable Energy Focus, 2011. **12**(2): p. 7.
128. OFGEM. *About the RO*. 2002; Available from: https://www.ofgem.gov.uk/environmental-programmes/ro/about-ro.
129. Government, U. *Renewable Transport Fuel Obligation*. 2012; Available from: https://www.gov.uk/guidance/renewable-transport-fuels-obligation.
130. OFGEM. *About the ECO scheme*. 2019; Available from: https://www.ofgem.gov.uk/environmental-programmes/eco/about-eco-scheme.
131. Government, U. *Building Regulation 2010*. 2010; Available from: http://www.legislation.gov.uk/uksi/2010/2214/contents/made.
132. Government, U. *Boiler Plus Regulation*. 2018; Available from: https://www.vaillant.co.uk/homeowners/advice-and-knowledge/boiler-plus/what-is-boiler-plus-1173513.html.
133. Government, U. *Conservation of fuel and power: Approved Document L*. 2014; Available from: https://www.gov.uk/government/publications/conservation-of-fuel-and-power-approved-document-l.
134. Government, U. *The Energy Efficiency (Private Rented Property) (England and Wales) Regulations 2015*. 2015; Available from: https://www.legislation.gov.uk/ukdsi/2015/9780111128350/contents.
135. Government, U., *Fuel and power (VAT Notice 701/19)*. 2016.
136. Government, U., *Heat Networks Delivery Unit*. 2017.
137. Deparment for Business, E.I.S., *HEAT NETWORKS INVESTMENT PROJECT*. 2018.
138. Government, C.a.L. *Code for Sustainable Homes Technical Guide*. 2010; Available from: https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/5976/code_for_sustainable_homes_techguide.pdf.
139. Government, C.a.L., *Cost of building to the Code for Sustainable Homes*. 2011.
140. House of Commons Housing, C.a.L.G.C., *Modern methods of construction*. 2017.
141. Gillott, M., L.T. Rodrigues, and C. Spataru. *Low-carbon housing design informed by research*. in *Proceedings of the Institution of Civil Engineers-Engineering Sustainability*. 2010. Thomas Telford Ltd.
142. Mazzeo, D., N. Matera, C. Cornaro, G. Oliveti, P. Romagnoni, and L. De Santoli, *EnergyPlus, IDA ICE and TRNSYS predictive simulation accuracy for building thermal behaviour evaluation by using an experimental campaign in solar test boxes with and without a PCM module.* Energy and Buildings, 2020: p. 109812.
143. U.S, B.T.O.o. *Department of Energy's (DOE) Building Technologies Office (BTO) EnergyPlus™ 9.2(2020)*. 2020.
144. Crawley, D.B., J.W. Hand, M. Kummert, and B.T. Griffith, *Contrasting the capabilities of building energy performance simulation programs.* Building and Environment, 2008. **43**(4): p. 661-673.

145. IES, V.E. *Integrated environmental solutions virtual environment*. 2020; Available from: https://www.iesve.com/.
146. Imam, S., D.A. Coley, and I. Walker, *The building performance gap: Are modellers literate?* Building Services Engineering Research and Technology, 2017. **38**(3): p. 351-375.
147. Al-janabi, A., M. Kavgic, A. Mohammadzadeh, and A. Azzouz, *Comparison of EnergyPlus and IES to model a complex university building using three scenarios: Free-floating, ideal air load system, and detailed.* Journal of Building Engineering, 2019. **22**: p. 262-280.
148. Trimble, *SketchUp Instructions* 2010.
149. Morton, A. *Temperature variations in UK heated homes*. 2012; Available from: http://www.lolo.ac.uk/wp-content/uploads/2016/02/1355492943_FinaldissertationAMorton.pdf.
150. Viessmann. *What is the ideal room temperature?* 2020; Available from: https://www.viessmann.co.uk/heating-advice/what-is-the-ideal-room-temperature.
151. Design, C.G.A.E., *The Chartered Institution of Building Services Engineers*. 2006, London.
152. Guide, C. and B. Volume, *Installation and equipment data.* CIBSE, London, 1986.
153. CIPE. *Hot Water Scalds*. 2020; Available from: https://www.ciphe.org.uk/consumer/safe-water-campaign/hot-water-scalds/.
154. TMVA. *Thermostatic Mixing Valve Manufactrurers Association*. 2020; Available from: file:///C:/Users/ezxwz/Downloads/Recommended%20Code%20of%20Practice%20for%20Safe%20Water%20Temperatures.pdf.
155. Ministry of Housing, C.L.G., *Building regulation in England setting standards for the energy performance of new and existing buildings*. 2018.
156. Schiano-Phan, R., B. Ford, M. Gillott, and L. RODRIGUES, *The Passivhaus standard in the UK: Is it desirable.* Is it achievable, 2008.
157. Loveday, D.L., K. Vadodaria, V. Haines, N. Hewitt, T. Hyde, P. Griffiths, B. Critoph, P.C. Eames, P. Banfill, and M. Gillott, *Refurbishing the UK's' hard to treat'dwelling stock: Understanding challenges and constraints.* 2011.
158. Saint-Gobain, 2010.
159. Vadodaria, K., D. Loveday, S. Hodder, M. Gillot, and C. Spataru. *Energy efficient retrofit technologies for existing homes and its impact on thermal comfort-initial findings from the CALEBRE project*. in *Clima 2010 Congress*. 2010.
160. Hormazabal, N., M. Gillott, G. Guzman, and G. Revell. *The Effect of Technological User Control Systems on Occupants of Sustainable Energy Homes*. in *26th Conference on Passive and Low Energy Architecture*.
161. Spataru, C., M. Gillott, and M.R. Hall, *Domestic energy and occupancy: a novel post-occupancy evaluation study.* International Journal of Low-Carbon Technologies, 2010. **5**(3): p. 148-157.
162. Spataru, C. and M. Gillott, *The use of intelligent systems for monitoring energy use and occupancy in existing homes*, in *Sustainability in Energy and Buildings*. 2011, Springer. p. 247-256.
163. Banfill, P.F., S. Simpson, M.C. Gillott, and J. White, *The potential for energy saving in existing solid wall dwellings through mechanical ventilation and heat recovery.* European Council for an Energy Efficient Economy, summer study, 2011: p. 6-031.
164. Banfill, P.F., S.A. Simpson, M.C. Gillott, and J. White. *Mechanical ventilation and heat recovery for low carbon retrofitting in dwellings*. in *World Renewable Energy Congress-Sweden; 8-13 May; 2011; Linköping; Sweden*. 2011. Linköping University Electronic Press.
165. CIBSE, E.D., *CIBSE Guide a, Chart.* Inst. Build. Serv. Eng. London, 2015.
166. Paseka, S., Z. Kapelan, and D. Marton, *Multi-Objective Optimization of Resilient Design of the Multipurpose Reservoir in Conditions of Uncertain Climate Change.* Water, 2018. **10**(9): p. 1110.

167. Rahman, A., N. Fumo, and A.D. Smith. *Simplified modeling of thermal storage tank for distributed energy heat recovery applications*. in *ASME 2015 9th International Conference on Energy Sustainability collocated with the ASME 2015 Power Conference, the ASME 2015 13th International Conference on Fuel Cell Science, Engineering and Technology, and the ASME 2015 Nuclear Forum*. 2015. American Society of Mechanical Engineers Digital Collection.
168. Rodríguez-Hidalgo, M.C., P.A. Rodríguez-Aumente, A. Lecuona, M. Legrand, and R. Ventas, *Domestic hot water consumption vs. solar thermal energy storage: The optimum size of the storage tank.* Applied Energy, 2012. **97**: p. 897-906.
169. Solar, V. *V15 V20 V30 Roof Integrated Solar Heating Panels*. 2018; Available from: http://www.viridiansolar.co.uk/assets/files/Clearline-Solar-Thermal-Panel-Datasheet.pdf.
170. Hoval. *EnerVal (200-10000) Thermal Storage Vessels*. 2020; Available from: https://pdf.archiexpo.com/pdf/hoval-italia/enerval-200-10000-thermal-storage-vessels/66161-240894.html.
171. Airtricity, S. *How do the rates work?* 2019; Available from: https://www.sseairtricity.com/uk/home/help-centre/understanding-your-bill/rates-and-charges/how-do-the-rates-work/.
172. Wood, A.J. and B. Wollenberg, *Power Generation, Operation, and Control, New York, NY: John Wiley&Sons*. 1996, Inc.
173. BREBE. *Advice Notes on Solar Thermal Technology Economics for the NPA Region*. 2017; Available from: http://grebeproject.eu/wp-content/uploads/2017/10/GREBE-Advice-Notes-SOLAR-Thermal.pdf.
174. Panapakidis, D., *Solar water heating systems study reliability, quantitative survey and life cycle cost method.* Master of Science Thesis, department of mechanical engineering, University of Strathclyde, Glasgow, 2009.
175. ENERGY.GOV. *Solar Water Heating System Maintenance and Repair*. 2019; Available from: https://www.energy.gov/energysaver/solar-water-heating-system-maintenance-and-repair.
176. ETA. *ETA HACK 20 - 200 kW A new standard for safe fuel conveying systems*. 2020; Available from: http://lintonandrobinson.co.uk/wp-content/uploads/2017/06/ETA-HACK-20-200-Woodchip-Boiler.pdf.
177. BOSCH. *Greenstar Ri (9kW to 24kW) overview*. 2019; Available from: https://www.worcester-bosch.co.uk/products/boilers/directory/greenstar-ri-9kw-to-24kw.
178. Rheem. *COMMERCIAL HEAT PUMP*. 2015; Available from: https://assets.ctfassets.net/phagqs82lusw/5BTr9WkMMwagyUgyGaaeEM/70545b6ce368b7cec5a3a9e5047f7b22/Heat_Pump_-_Commercial_brochure_Sept_2015.pdf.
179. Hawkes, A.D. and M.A. Leach, *Modelling high level system design and unit commitment for a microgrid.* Applied Energy, 2009. **86**(7): p. 1253-1265.
180. ETA. *The pellet boiler for pleasantly warm single-family homes*. 2019; Available from: https://www.eta.co.at/index.php?id=91&download=75dd09a9f11a340f7d02a6b85a7af174.
181. BOSCH. *Greenstar CDi Highflow overview*. 2016 [cited 2020; Available from: https://www.worcester-bosch.co.uk/products/boilers/directory/greenstar-cdi-highflow.
182. GOV.UK. *Carbon Emissions Tax*. 2018 [cited 2019; Available from: https://www.gov.uk/government/publications/carbon-emmisions-tax/carbon-emmisions-tax.
183. Trust, E.S. *Fuel prices and carbon intensity*. 2018; Available from: https://www.energysavingtrust.org.uk/about-us/our-calculations.
184. Change, U.D.o.E.C. *Wood chip compares favourably with fossil fuels*. 2017; Available from: https://www.forestry.gov.uk/forestry/infd-9peg3u.
185. Strehler, A., *Technologies of wood combustion.* Ecological Engineering, 2000. **16**: p. 25-40.
186. Chau, J., T. Sowlati, S. Sokhansanj, F. Preto, S. Melin, and X. Bi, *Techno-economic analysis of wood biomass boilers for the greenhouse industry.* Applied Energy, 2009. **86**(3): p. 364-371.

APPENDIX A The models for CEHs in SketchUp

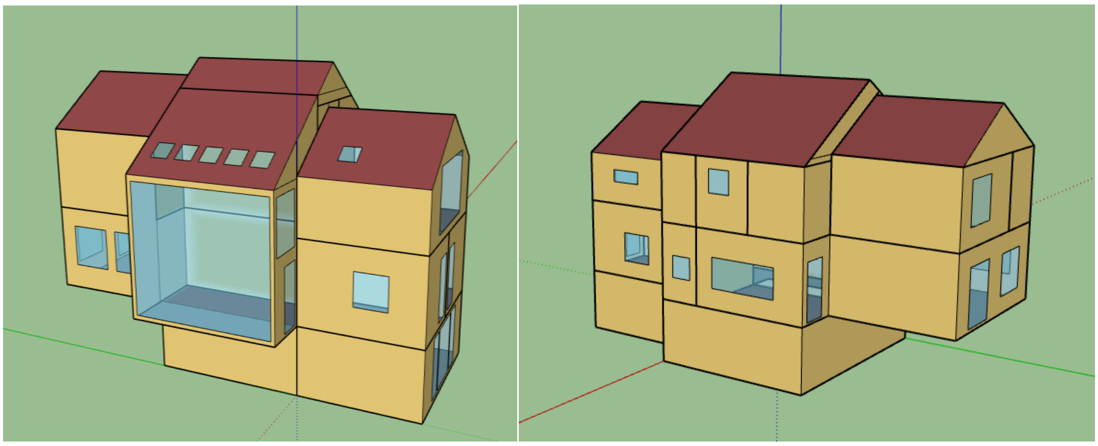


Figure A-1 Mark group house model

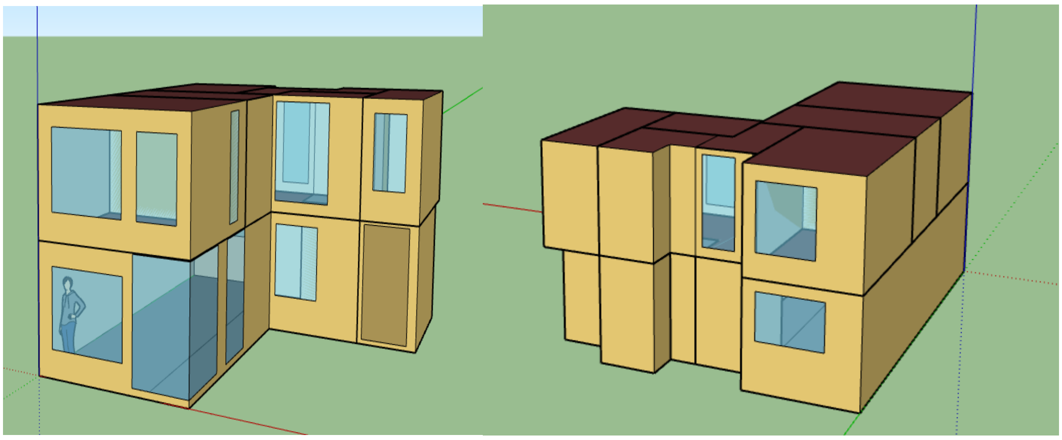


Figure A-2 Nottingham house model

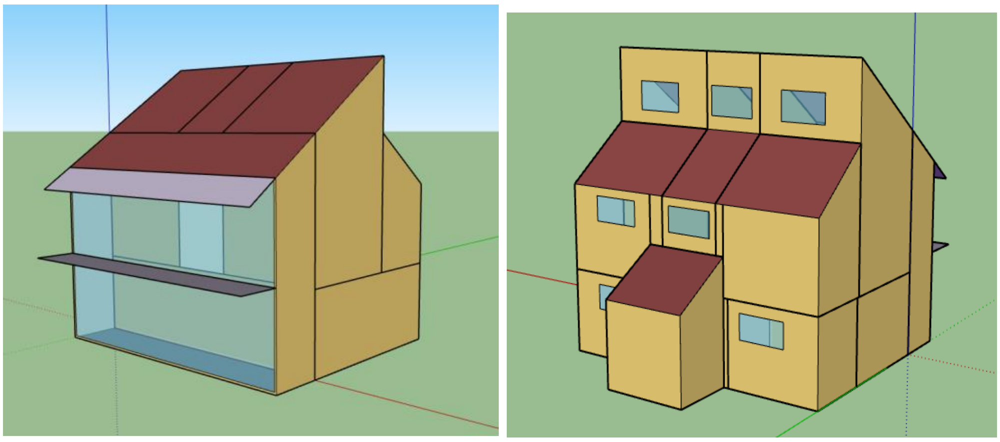


Figure A-3 BASF house model

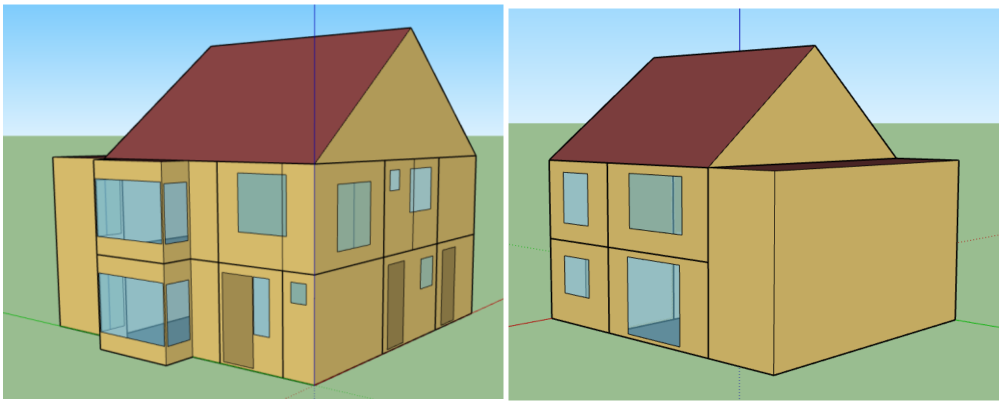


Figure A-4 Eon house model

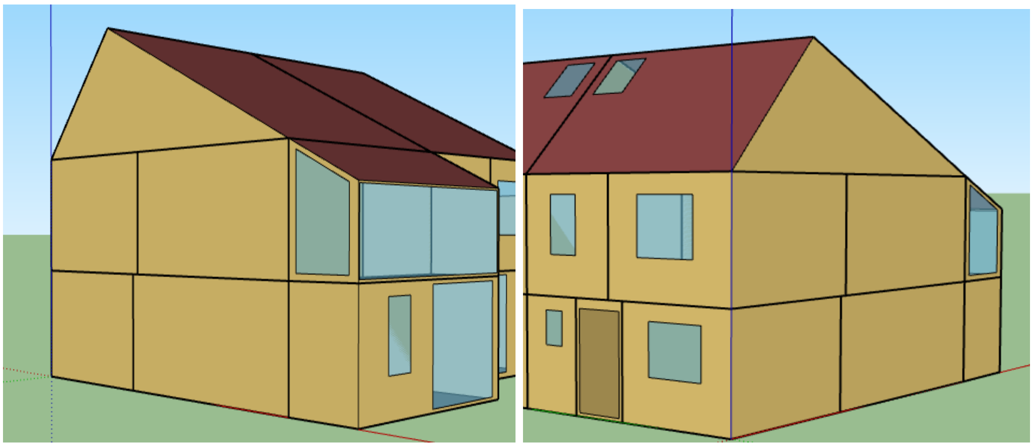


Figure A-5 Tarmac 4 house model

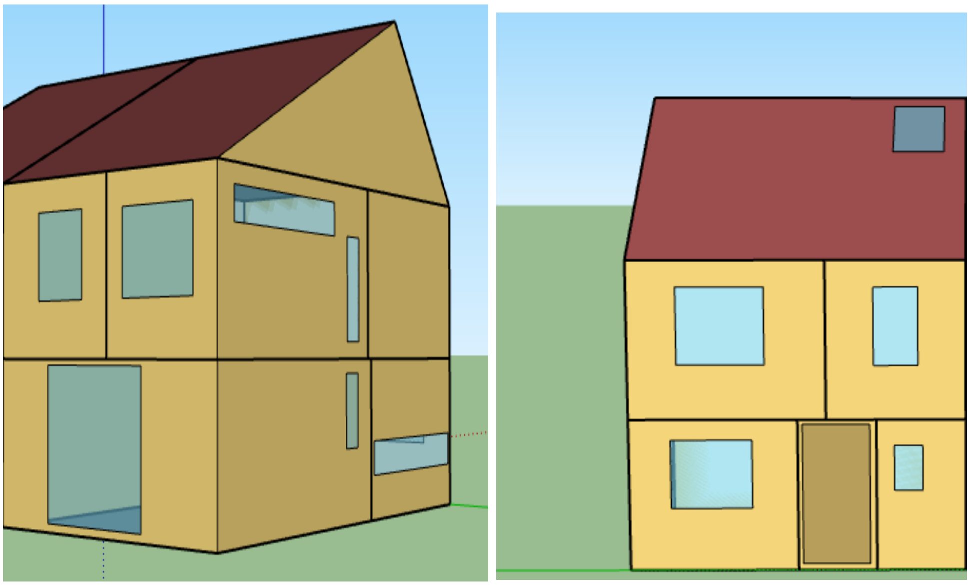


Figure A-6 Tarmac 6 house model

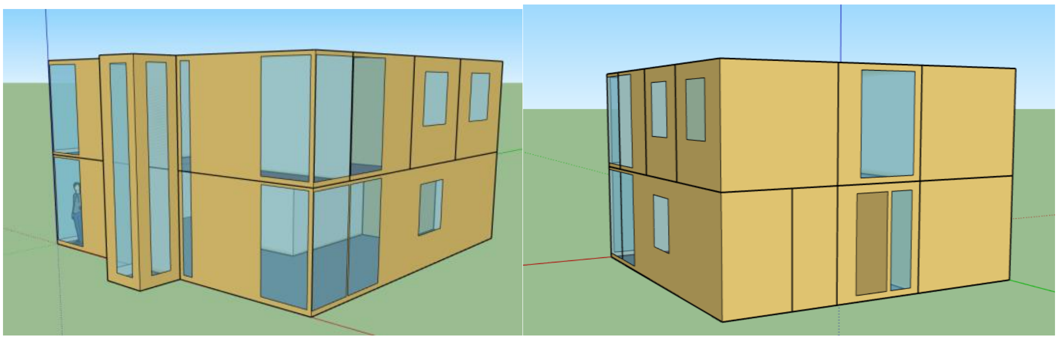


Figure A-7 David Wilson model

APPENDIX B MATLAB model: simulation of LTDH with multiple heat sources

```
n=3.1; %collector area
Q_solar = s1*n; % hourly collector solar energy during a year
 initialtank=5; %initial thermal store capacity
  Q_load=h; % hourly heat load during a year
 interval=24;
 T_tol= interval*365;% total simulation time in hour
 Q_hp=10; % output of heat pump
 Q_bio=5; % output of biomass boiler
 Q_gas=5;% output of biomass boiler
  Q_storage= zeros(T_tol,1);% storage capacity
   t=zeros(interval,1)% each day time
   if Q_hp>=6
    Q_storage(1)= Q_solar(1)+ initialtank+Q_hp-Q_load(1);
   else
       Q_storage(1)=Q_solar(1)+ initialtank+Q_hp+Q_bio+-Q_load(1);
   end

 for i=2:T_tol;
     if i <= 3624; % januaray to may
        t= mod(i,interval);
        if t<=7 & t>0 % time from 1 to 7am
            if Q_storage(i-1)+Q_solar(i)-Q_load(i)< initialtank
            Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i);
             if Q_storage(i)< initialtank
                 Q_storage(i)=Q_storage(i)+Q_bio;
                 if Q_storage(i)< initialtank
                      Q_storage(i)=Q_storage(i)+Q_gas;
                 end
             end

            else
                 Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i);
            end
        else
            if Q_storage(i-1)+Q_solar(i)-Q_load(i)< initialtank
               Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_bio-
Q_load(i);
               if Q_storage(i)< initialtank
               Q_storage(i)=Q_storage(i)+Q_hp;
                    if Q_storage(i)< initialtank;
                         Q_storage(i)=Q_storage(i)+Q_gas;
                    end
               end

            else
                 Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i);
            end
        end
     end

         if i >3624 & i<= 6552; % june to sepetember
              if Q_storage(i-1)+Q_solar(i)-Q_load(i)< initialtank
            Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i);
             if Q_storage(i)< initialtank
```

```
                    Q_storage(i)=Q_storage(i)+Q_bio;
                    if Q_storage(i)< initialtank
                         Q_storage(i)=Q_storage(i)+Q_gas;
                    end
                end
                 else
                      Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i);

                 end
          end
        if i >6552; % october to december
          t= mod(i,interval);
          if t<=7& t>0% time from 1 to 7am
               if Q_storage(i-1)+Q_solar(i)-Q_load(i)< initialtank
               Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i);
                if Q_storage(i)< initialtank
                    Q_storage(i)=Q_storage(i)+Q_bio;
                    if Q_storage(i)< initialtank
                         Q_storage(i)=Q_storage(i)+Q_gas;
                    end
                end

               else
                    Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i);
               end
          else
               if Q_storage(i-1)+Q_solar(i)-Q_load(i)< initialtank
                  Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_bio-
Q_load(i);
                  if Q_storage(i)< initialtank
                  Q_storage(i)=Q_storage(i)+Q_hp;
                       if Q_storage(i)< initialtank;
                            Q_storage(i)=Q_storage(i)+Q_gas;
                       end
                  end

               else
                    Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i);
               end
          end
      end
  end

        [x,y]=find(Q_storage==max(Q_storage(:)))
        T=1:T_tol;
        plot(T,Q_storage(1:T_tol))

      N = sum(Q_storage<5)
      Q_bigstorage=max(Q_storage)
```

APPENDIX C MATLAB model: simulation thermal storage operation

```
n=3.1; %collector area
Q_solar = s1*n; % hourly collector solar energy during a year
 initialtank=5; %initial thermal store capacity
  Q_load=h; % hourly heat load during a year
  t_ambient=tambient;% the ambient temperature
 interval=24;
 T_tol= interval*365;% total simulation time in hour
  Q_hp=10; % output of heat pump
 Q_bio=15; % output of biomass boiler
 Q_gas=20;% output of biomass boiler
  Q_storage= zeros(T_tol,1);% storage capacity
  Q_loss=zeros(T_tol,1);% heat loss of storage
   t=zeros(interval,1)% each day time
   U=0.34;% the heat transfer coefficient
   A=7.73;% is the surface area of storage capacity
   t_m=52.5;% the mean temperature of thermal store
   Q_loss=U*A*(t_m-t_ambient)/1000;% the heat loss of thermal store
      if Q_hp>=6
    Q_storage(1)= Q_solar(1)+ initialtank+Q_hp-Q_load(1)-Q_loss(1);
   else
       Q_storage(1)=Q_solar(1)+ initialtank+Q_hp+Q_bio+-Q_load(1)-
Q_loss(1);
   end

  for i=2:T_tol;
      if i <= 3624; % januaray to may
         t= mod(i,interval);
         if t<=7 & t>0 % time from 1 to 7am
             if Q_storage(i-1)+Q_solar(i)-Q_load(i)-Q_loss(i)<
initialtank
             Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i)-
Q_loss(i);
              if Q_storage(i)< initialtank
                  Q_storage(i)=Q_storage(i)+Q_bio;
                  if Q_storage(i)< initialtank
                       Q_storage(i)=Q_storage(i)+Q_gas;
                  end
              end

             else
                  Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i)-
Q_loss(i);
             end
         else
             if Q_storage(i-1)+Q_solar(i)-Q_load(i)-Q_loss(i)<
initialtank
                Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_bio-
Q_load(i)-Q_loss(i);
                if Q_storage(i)< initialtank
                Q_storage(i)=Q_storage(i)+Q_hp;
                     if Q_storage(i)< initialtank;
                          Q_storage(i)=Q_storage(i)+Q_gas;
                     end
```

```
                     end

                  else
                       Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i)-
Q_loss(i);
                  end
              end
           end

               if i >3624 & i<= 6552; % june to sepetember
                    if Q_storage(i-1)+Q_solar(i)-Q_load(i)-Q_loss(i)<
initialtank
                  Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i)-
Q_loss(i);
                   if Q_storage(i)< initialtank
                       Q_storage(i)=Q_storage(i)+Q_bio;
                       if Q_storage(i)< initialtank
                            Q_storage(i)=Q_storage(i)+Q_gas;
                       end
                   end
                    else
                         Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i)-
Q_loss(i);

                    end
              end
            if i >6552; % october to december
              t= mod(i,interval);
              if t<=7& t>0% time from 1 to 7am
                  if Q_storage(i-1)+Q_solar(i)-Q_load(i)-Q_loss(i)<
initialtank
                  Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_hp-Q_load(i)-
Q_loss(i);
                   if Q_storage(i)< initialtank
                       Q_storage(i)=Q_storage(i)+Q_bio;
                       if Q_storage(i)< initialtank
                            Q_storage(i)=Q_storage(i)+Q_gas;
                       end
                   end

                  else
                       Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i)-
Q_loss(i);
                  end
              else
                  if Q_storage(i-1)+Q_solar(i)-Q_load(i)-Q_loss(i)<
initialtank
                     Q_storage(i)= Q_solar(i)+ Q_storage(i-1)+Q_bio-
Q_load(i)-Q_loss(i);
                     if Q_storage(i)< initialtank
                     Q_storage(i)=Q_storage(i)+Q_hp;
                          if Q_storage(i)< initialtank;
                               Q_storage(i)=Q_storage(i)+Q_gas;
                          end
                     end

                  else
                       Q_storage(i)=Q_solar(i)+Q_storage(i-1)-Q_load(i)-
Q_loss(i);
                  end
```

```
        end
    end
end
```